AD

TARDEC No. TR13669

RESEARCH NEEDED FOR MORE COMPACT INTERMITTENT COMBUSTION PROPULSION SYSTEMS FOR ARMY COMBAT VEHICLES

VOLUME II
Appendices

INTERIM REPORT
TFLRF No. 296

By
Blue Ribbon Committee (BRC)

Prepared for
U.S. Army Tank-automotive and Armaments Command (TACOM)
U.S. Army Tank-Automotive Research, Development and Engineering Center (TARDEC)
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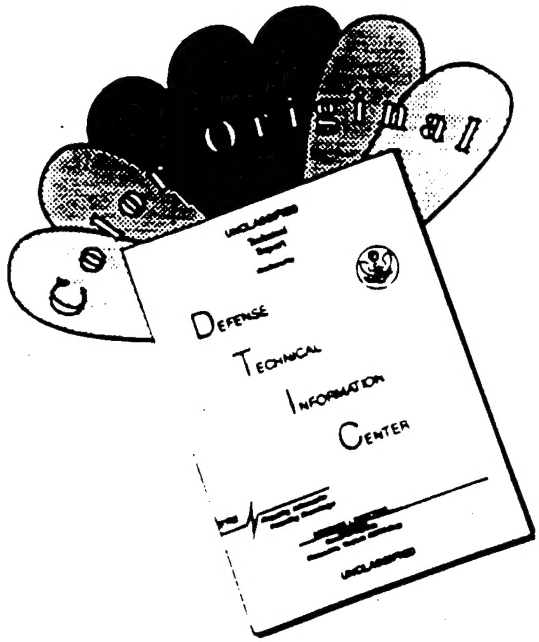

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13. ABSTRACT (Maximum 200 words)

Because of the unique performance requirements of combat vehicles, it is not possible to achieve the required propulsion system power density by adaptation of commercial engines; the **Army** must ensure that the research and development required to achieve the needed propulsion system power density is accomplished.

(continued on back)

14. SUBJECT TERMS

Combustion	Vehicle performance	Combat vehicle - survivability
Propulsion systems	Specific power	Mobility systems
Intermittent combustion engines	Fuels and lubricants	Electric drive
Engine subsystems	Combat vehicle - performance	Mechanical transmissions
Propulsion system power density		

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To assist TACOM in identifying the research and development needed in the next decade for intermittent combustion combat engines, a BRC was established. Assuming the BRC recommended research is successfully accomplished, propulsion system volume could be reduced for the same power output, or for the same system volume, vehicle power could be markedly increased. The BRC identified hp/ton as a significant vehicle performance factor. By inserting advanced propulsion systems AIPS, or BRC, in the **existing system** volume of five different vehicles, the BRC study shows that hp/ton increases are dramatic. For example, if development of the AIPS were completed, hp/ton for the AGT1500 hp M1 would increase from 22.2 hp/ton to 35 hp/ton using AIPS technology; use of BRC technology would result in 43.3 hp/ton. **For new vehicles**, again assuming completion of AIPS development, a weight decrease of 7 tons from a nominal 60-ton AGT1500 vehicle could be achieved; an additional 3 tons could be achieved using BRC technology.

TABLE OF CONTENTS — VOLUME II
Appendices

TABLE OF CONTENTS — VOLUME II, CONT'D

TABLE OF CONTENTS — VOLUME II, CONT'D

TABLE OF CONTENTS — VOLUME II, CONT'D

TABLE OF CONTENTS — VOLUME II, CONT'D

Note: Appendices in both Volume I (Arabic numerals) and Volume II (Roman numerals) correspond with like numbered sections in the Volume I narrative, to facilitate the location of common report information and maintain continuity. Hence, certain unnecessary numbers in the appendices have not been used. Volume II expands selected sections of Volume I.

LIST OF ILLUSTRATIONS — VOLUME II

LIST OF ILLUSTRATIONS — VOLUME II, CONT'D

LIST OF TABLES — VOLUME II

GLOSSARY — VOLUME II

A-A	Air-to-Air (charge air cooling)
AC	Alternating Current
AGS	Armored Gun System
AGT	Army Gas Turbine
AIPS	Advanced Integrated Propulsion System
APU	Auxiliary Power Unit
ASM	Armored System Modernization
ATAC	Army Tank-Automotive Center
AVCR	Air-Cooled Variable Compression Ratio (diesel engine)
b	Constant volume portion of total combustion
B × S	Bore × Stroke
BFD	Battlefield Day (duty cycle)
bhp	Brake horsepower
BMEP	Brake Mean Effective Pressure
BSAC	Brake Specific Air Consumption
BRC	Blue Ribbon Committee
BSFC	Brake Specific Fuel Consumption
Btu	British thermal unit
cfm	cubic feet per minute
CID	Cubic Inch Displacement
CONUS	Continental United States
COSCOM	Corps Support Command
CR	Compression Ratio
CV	Compression-Ignited Vee (engine)
dAIPS	diesel Advanced Integrated Propulsion System
DC	Direct Current
DECT	Diesel Engine Component Technology
DF	Diesel Fuel
DF-A	Diesel Fuel-Arctic
ECI	Electronic Controlled Injection
ECU	Electronic Control Unit
ε_i	Aftercooler effectiveness
EM	Electromagnetic (gun)
Epump	Pumping Efficiency
ETC	Electrothermal Cannon
EV	Engine Volume
EW	Engine Weight
F/A	Fuel/Air Ratio
fpm	feet per minute
FTPT	Footprint
ghp	gross horsepower
GIMEP	Gross Indicated Mean Effective Pressure
gph	gallons per hour
GVW	Gross Vehicle Weight

GLOSSARY — VOLUME II, CONT'D

hp	horsepower
HPD	Higher Power Density
HPDPS	Higher Power Density Propulsion System
ICE	Internal Combustion Engine
IGBT	Insulated Gate Bipolar Transistor
IMEP	Indicated Mean Effective Pressure
In-CONUS	In the continental United States
JP	Jet Fuel (petroleum)
kph	kilometers per hour
Ksi	Thousand pounds per square inch
LCTE	Lowest Comparable Technology Estimate
LER	Loss Exchange Ratio
LHV	Lower Heating Value
LOTS	Logistics Over the Shore
M_a/M_f	Mass of Air per Mass of Fuel
MIL SPEC	Military Specification
mpg	miles per gallon
MTCB	Mobility Technology Center-Belvoir
MTMC	Military Traffic Management Command
MTU	Motor Turbinen Union (Germany)
η	revolutions per minute
NA	Naturally Aspirated
NATO	North American Treaty Organization
NBC	Nuclear, Biological, and Chemical
Pmax	Maximum Pressure
PA	Peacetime Annual (duty cycle)
PCU	Power Conversion Unit
PM	Permanent Magnet
PMEP	Pumping Mean Effective Pressure
pph	pounds per hour
PR	Pressure Ratio
psia	Pounds per square inch, absolute
PV	Pressure Volume
q	Cooling coefficient
ρ/ρ_o	Inlet air density ratio
RAMEP	Rubbing and Accessory Mean Effective Pressure
rpm	revolutions per minute
SwRI	Southwest Research Institute
T	Torque
TACOM	U.S. Army Tank-automotive and Armaments Command
tAIPS	turbine Advanced Integrated Propulsion System
TARDEC	U.S. Army Tank-Automotive Research, Development and Engineering Center
TE	Tractive Effort

GLOSSARY — VOLUME II, CONT'D

Tgw	Gas Side Wall Temperature
TMEPS	Transverse Mounted Engine Propulsion System
TOE	Table of Organization and Equipment
TRR	Top Ring Reversal (temperature)
Ttrg	Top ring groove temperature
VIT	Variable Inlet Timing (valve)

VOLUME II

Appendices

APPENDIX II

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(Note: Appendix I is not used.)

APPENDIX II

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APPENDIX III

The Need for Higher Power Density (HPD) Propulsion Systems for Future Combat Vehicles

APPENDIX III

THE NEED FOR HIGHER POWER DENSITY (HPD) PROPULSION SYSTEMS FOR FUTURE COMBAT VEHICLES

The Demand for Smaller, Lighter Vehicles

The reformulation of national military strategy in the wake of the end of the Cold War, plus recent military events, has led to the consideration of the impact of the new strategy on the design of the combat vehicles used to implement that strategy. This, in turn, influences the requirements upon the propulsion system and in turn, its design. However, the overall assessment of the military worth (i.e., war-winning capability) of one vehicle design versus another is difficult to fully quantify. This is due in large part to the inadequacy of existing war gaming scenarios which fail in two ways. First, they are insensitive to the details of vehicles generally not being able, for instance, to distinguish between different terrain capabilities (while most could be reprogrammed, it is not a menu option). Second, they do not respond to major capability improvements which cannot be fully exploited in the absence of totally new tactics. In this latter case, the multiple running of games involving live decision makers should, over time, lead to some recognition of the possibilities. Nonetheless, certain trends are established that give credence, if not proof, to the arguments presented below.

Battles are fought with organized and equipped forces, not by vehicles per se. Thus, while the main battle tank is the centerpiece and the raison d'etre of an armored division, the 348 tanks represent but a fraction of the firepower available (TABLE III-1). By doctrine and common sense, they cannot fight alone but must operate in concert with other complementary weapons and support.

TABLE III-1. Typical Armored Division (6 × 4 × 2)*

Item	Number	Short Tons
Tracked vehicles	1,895	51,352
Trucks	3,031	23,913
Trailers	1,627	4,206
Aircraft	127	566
Equipment	--	5,600
Subtotal		**85,637**
30-day sustainment		**104,970**

* 17,000 personnel, TOE 87000J430, 6 armored battalions, 4 infantry battalions, 2 aviation battalions

Although an Army's presence may be merely to deter, its principal function (and indeed, the threat behind its deterence capability) is to close with and destroy the enemy. The notion of "closing with" entails the military concept maneuver, while "destroy" entails mass and fire. This in turn requires action at three recognized levels: Strategic, Operational, and Tactical.

III.1 Strategic

"Strategic" entails the movement between the home and the theater of operations. The essential requirement at this level is the ability to deploy the force in a timely fashion and to sustain it thereafter. In the Eurocentric cold war mentality during which the single foe was the Soviet Union, the deployment issue was addressed by forward deployment to the presumed theater of operations (Europe) with a massive logistics capability built up in-theater. Reinforcement and continued sustainment involved fast sealift (transatlantic only) with required Naval defense of the sea lanes of communication. In the new multipolar political world, the Army is generally in the U.S. and may be deployed nearly anywhere--so far, Panama, Kuwait, Somalia, and Bosnia--for disparate reasons, with vastly different intensities of combat, with a wide range of host nation support, and with great environmental extremes.

Timely arrival, despite possibly great distances and large forces, is mandatory. At best, the real threat of a U.S. defense of its interests will stop hostilities before they start; at worst, the U.S. would have to fight its way ashore at prohibitive expense of personnel and materiel.

Deployment has three generally recognized stages, each with its own transportation problems and time scales of operation. These are in-CONUS, movement from embarkation to the theater, and in-theater movement to operational positions.

III.1.1 In-CONUS

In-CONUS movement, i.e., from post to embarkation point (sea or air), is accomplished by highway and rail. Despite the loading time, rail is considered simpler because of the impediments promulgated by the various state highway departments on the highway. Tracked vehicles, even light ones within state weight limits, will be moved by trailers. However, the ability to use commercial tractor/trailer rigs to haul them is greatly enhanced by reduced weight. Rail is somewhat constrained, like the highways, by bridge weight limitations, thus requiring occasional detours. The railcars themselves generally have no problem with the cargo weight.

III.1.2 Movement from embarkation to the theater

Movement from embarkation to the theater is by sea or air, naturally. There is much ado about airlift now. It is very utilitarian in the deployment of a small light force for policing action or threat response ("drawing a line in the sand"), but for the deployment of heavy forces (even those which may be lightened through advanced technology, though with substantial heavy firepower), airlift is a non-starter. For instance, as TABLE III-2 shows, the 82nd AB is about one-fifth the weight of an armored division (24th Mech ID is close), yet in Operation Desert Shield where, in fact, speed was essential to draw that "line in the sand," only 50 percent of that lightest of all divisions was transported by air! The primary reason being, as shown by TABLE III-3, that there is considerable competition for airlift capability. The Air Force itself demands much in order

to deploy the ground support equipment for its initial fighter squadrons (one of the best initial defenses) and to deploy the equipment and units for the unloading and warehousing of further units deploying by air.

TABLE III-2. XVIII Corps Breakdown

Unit	Short Tons
82 AB	21,943
101 AA	32,547
3 ACR	30,763
24 ID (M)	95,020
Total CBT Maneuver	180,273
COSCOM	360,546
TOTAL	**540,819**

TABLE III-3. Air Deployment Experience
Desert Shield Sorties (C+0 to C+23)

Cargo	C-141	C-5
Army		
82nd* (50 percent of unit)	244	100
101st* (10 percent of unit)	49	41
24th	10	2
Total Army	303 (30 percent)	143 (35 percent)
USAF	197	92
USMC	162	56
Navy	3	7
Centcom	9	2
Other**	325 (33 percent)	105 (26 percent)
Total sorties	**999**	**405**

* Total 82nd = 21,943 ston
Total 101st = 32,547 ston
** Unidentified, resupply, etc.

TABLE III-3 also points out the fallacy of talking in terms of deploying divisions. Corps are deployed. Firstly, it is the Corps that orchestrates the operational (theater) level maneuver. Secondly, it is the Corps that has the equipment to support the individual maneuver units (divisions). Note that for the XVIII Corps (the allegedly "light" Contingency Corps), the Corps Support Command (COSCOM) outweighs the fighting arm by 2:1. The total Corps weight exceeds that of five aircraft carriers, for comparison.

Nonetheless, if one is to deploy by air, no matter how small a fraction of the force, weight does count. The Staff Officers' Field Manual, FM 101-10-1/2 advises that only in the movement of the heaviest units (e.g., a tank battalion) does the average cargo load--a mix of heavy and light equipment--hit the aircraft weight limit at about the same time that the floor space is consumed. Lighter units (including future "heavy" units), will fill the space, on average, before the aircraft weight limit is reached. This would suggest that small size rather than weight should drive design. However, specifically aiming for lower weight will allow more fuel on the aircraft and hence, more range or fewer air-to-air refueling, a burdensome logistics task.

[As an aside, the above remarks point out the fallacy of arguing in favor of the ability to squeeze X or Y numbers of a specific type of vehicle into a single airframe. If another M2, for instance, cannot be put into a C5 because the weight limit would be exceeded, there are plenty of other pieces of miscellaneous equipment belonging to the deploying unit that can be stuffed into the aircraft instead. There are enough aircraft sorties necessary to distribute the heavy equipment.]

Sealift, including prepositioned equipment afloat (a form of forward deployment), is the most realistic, if slower, approach to deployment of large and heavy forces. With sealift, deployment is NOT weight limited but strongly footprint limited. For instance, the fleet of eight (now) Fast Sealift ships intended for the deployment of the 24 ID (Mech), the heavy arm of the XVIII Corps, have a total of 1,640,000 ft^2 of deckspace ["Vehicle Characteristics for Shiploading," Military Traffic Management Command (MTMC)] of which 1,150,000 ft^2 is usable [packing factor of 0.7 as provided by MTMC (MTMC-TEA Rpt 0A 90-4f-22)]. This carries about 80 percent of the division (its footprint is 1,380,000 ft^2 per MTMC correspondence) or equivalently, 79,000 of its 95,000 stons (TABLE III-2). The carrying capacity of these eight ships, by comparison, is 28,000 stons each or 224,000 stons altogether. Thus, the fleet is loaded to only one-third of its weight capacity. Herein is a strong argument for reducing vehicle size, although the argument can be overstated. Combat vehicles account for only 385,000 ft^2 of the total footprint (TABLE III-4), and not all can equally be reduced in size (e.g., the M113 APC family is already quite petite and light). With disciplined design across the board (combat and tactical vehicles), the ALCV Thrust 5 Task Force concluded that a reduction of the divisional footprint of 23 percent was possible (TABLE III-5). The savings (70 percent), though, was largely due to the elimination of excess tactical vehicles (including trailers) that resulted from the economies gained through (a) the fuel savings attached to the lighter combat vehicles and fewer trucks, (b) the reduced maintenance support thereof, and (c) the reduced support requirements incumbent to the reduced crew sizes, a necessary feature in shrinking the vehicles.

Another sealift feature often ignored is Logistics Over the Shore (LOTS): the debarkation requirements (units and supplies) at undeveloped, shallow, or hostile ports. This requires the use

TABLE III-4. Tracked Vehicle Breakdown

Item	Number	Short Tons	ft^2
M113 Family*	876	9,585	123,818
M109 Family**	144	3,232	31,407
M2 Family†	325	9,347	72,221
M1	348	21,750	103,078
M88	86	4,764	26,221
Other‡	116	2,674	28,326
TOTAL	**1,895**	**51,352**	**385,071**

* M113, M577, M548, M981, M901, M106
** M109, M992
† M2, M3, M993
‡ M728, M9, M60, AVLB, M578

of convenient lighterage (barges). Herein, size and weight of vehicles are potentially of concern, for individual lighterage loads may be less than 100 stons.

Lastly, sealift will be largely responsible for the sustainment loads. In moderately intense combat, a heavy division will consume its own weight in fuel (60 percent), ammunition (30 percent), and spares and miscellaneous (10 percent) each 30 days (TABLE III-6). Although such intense consumption is unlikely to continue for extended durations (as determined by the logistics capabilities of the offensive army), considerable stockpiling early on is practiced. Sealift, recall, must also sustain the bombing consumption of the Air Force, which in Desert Storm dropped some 90,000 tons of bombs. In special theaters, such as the Middle East, where fuel is available indigenously, the strategic resupply of fuel may not be of such a consideration. Elsewhere, it may be. The fact remains that fuel consumption is essentially directly proportional to the weight of the Army. As the combat vehicle force of an armored division constitutes 56 percent of the total weight of the division and consumes 73 percent of the fuel, economies in combat vehicle weight profoundly impact the logistics burden, at least at the Division level, although this may be seen as more of an operational consideration than a strategic one.

III.1.3 In-theater movement to operational positions

In-theater movement is highly varied, depending on the theater in question. It will range from the tropical mountainous regions (e.g., Panama) to dry, flat desert (e.g., Saudi Arabia, Iraq, Kuwait) and in the future, perhaps other extremes of terrain, soil, and infrastructure, such as urban or arctic regions. Although a quantified war-fighting impact cannot be stated, even if the theater were postulated, it is apparent that lighter forces (in weight, not necessarily in firepower

TABLE III-5. Potential Reduction in Divisional Footprint

	A	A1	B	B1	C	C1	D	D1	E	E1	F	F1
	Number		Weight/Vehicle		FTPT/Vehicle		Total Weight		Total Footprint		Pol	
Item	New	Baseline	New	Baseline	New	Baseline	New	Baseline	New	Baseline	New	Baseline
Combat Vehicles												
M1	348	348	30	62.5	215.3	296.2	10,440	21,750	74,924	103,078		
M2	216	216	17	29	144.4	222.2	3,672	6,264	31,190	47,995		
M3	100	100	17	28.8	144.4	222.2	1,700	2,880	14,440	22,220		
M109	72	72	17	23.2	144.4	206.4	1,224	1,670	10,397	14,861		
M993	9	9	17	22.5	144.4	222.9	153	203	1,300	2,006		
M901	48	48	10	11	130.6	140.5	480	528	6,269	6,744		
M106	66	66	10	9	130.6	151.7	660	594	8,620	10,012		
AD	30	30	17	20.9	144.4	222.2	510	627	4,332	6,666		
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Total	889	889					18,839	34,516	151,472	213,582		
Tracked Supply Vehicles												
M113	287	287	10	10.7	130.6	140.5	2,870	3,071	37,482	40,324		
M992	72	72	17	21.7	144.4	229.8	1,224	1,562	10,397	16,546		
M981	308	308	10	11	130.6	140.5	3,080	3,388	40,225	43,274		
M577	167	167	10	12	130.6	140.5	1,670	2,004	21,810	23,464		
M88	86	86	30	55.4	215.3	304.9	2,580	4,764	18,516	26,221		
AVLB	16	16	40	45.9	450	485.4	640	734	7,200	7,766		
Other	73	73	17	17.7	146	198.7	1,241	1,292	10,658	14,505		
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Total	1,009	1,009					13,305	16,816	146,288	172,100		
Total Tracked Vehicle	1,898	1,898					32,144	51,332	297,759	385,681	278.6	1324.1
							New/Base=62.6%		New/Base=77.2%		21.0%	

TABLE III-5. Potential Reduction in Divisional Footprint, Cont'd

	A	A1	B	B1	C	C1	D	D1	E	E1	F	F1
	Number		Weight/Vehicle		FTPT/Vehicle		Total Weight		Total Footprint		Pol	
Item	New	Baseline	New	Baseline	New	Baseline	New	Baseline	New	Baseline	New	Baseline
Trucks												
HMMWV	782	853	1.8	2.0	96.7	107.4	1,408	2,218	75,588	91,612		
Pickup	558	558	2	2.9	105.3	117	1,116	1,618	58,757	65,286		
2 × 1/2 ton	930	959	3.3	6.6	139.0	173.7	3,069	6,329	129,233	166,578		
5-ton	445	462	5.5	11	191.1	238.9	2,448	5,082	85,048	110,372		
5-ton Tractor	220	256	5.25	10.5	154.8	193.5	1,155	2,688	34,056	49,536		
Tanker	15	73	9.5	19	240.1	266.8	143	1,387	3,602	19,476		
HEMTT	142	240	10	19.5	240.1	266.8	1,420	4,680	34,097	64,032		
HET	12	24	10	19.9	154.8	193.5	120	478	1,858	4,644		
Contact Shops	32	40	3.5	5.5	126.6	140.7	112	220	4,052	5,628		
Other	184	184	8	16.1	184.2	230.3	1,472	2,962	33,900	42,375		
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Total	3,320	3,649					12,462	27,662	460,191	619,540	59.7	209.2
Oilers												
Vans	42	46	3.1	3.88	184.1	204.6	130	178	7,734	9,412		
Small	1,123	1,265	0.92	1.15	82.3	91.4	1,033	1,455	92,378	115,621		
Tanker	21	64	3.3	6.6	206.9	258.6	69	422	4,344	16,550		
Flatbeds	243	243	3.6	7.04	199.4	249.2	875	1,711	48,444	60,556		
Shops	157	164	3.15	3.5	122.5	135.1	494	574	19,230	22,320		
Generators	227	227	1.75	1.85	80.6	89.5	397	419	18,284	20,316		
M747	12	24	8	16	429.2	429.2	96	384	5,150	10,301		
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Total	1,825	2,033					3,094	5,143	195,585	255,095		

TABLE III-5. Potential Reduction in Divisional Footprint, Cont'd

	A	A1	B	B1	C	C1	D	D1	E	E1	F	F1
	Number		Weight/Vehicle		FTPT/Vehicle		Total Weight		Total Footprint		Pol	
Item	New	Baseline	New	Baseline	New	Baseline	New	Baseline	New	Baseline	New	Baseline
Craft												
UH-1	6	6	3.03	3.03	390	390	18	18	2,340	2,340		
AH-1	8	8	3.89	3.89	523.9	523.9	31	31	4,191	4,191		
AH-64	36	36	6.48	6.48	433.7	433.7	233	233	15,613	15,613		
UH-60	27	27	8.5	8.5	529.4	529.4	230	230	14,294	14,294		
OH-58	50	50	1.08	1.08	237.4	237.4	54	54	11,870	11,870		
Total	127	127					566	566	48,308	48,308	155.7	173.0
Equipment							7,007	7,007	70,264	70,264	111.0	111.0
Grand Total	7,170	7,707					55,273	91,711	1,072,088	1,378,869	605.0	1817.3
							New/Base=60.3%		New/Base=77.8%		33.3%	
Personnel												
Tkbns	2,580	3,312										
Infbns	3,136	3,376										
Divarty	2,206	2,516										
Maint	1,031	1,200										
Supply/Support	1,292	1,316										
Aviat	1,469	1,518										
Ada	565	625										
Her	2,847	3,164										
Total Personnel	15,126	17,027										

TABLE III-5. Potential Reduction in Divisional Footprint, Cont'd

	A	A1	B	B1	C	C1	D	D1	E	E1	F	F1
	Number		Weight/Vehicle		FTPT/Vehicle		Total Weight		Total Footprint		Pol	
Item	New	Baseline	New	Baseline	New	Baseline	New	Baseline	New	Baseline	New	Baseline
Daily Consumables												
Pol	605	1,817										
Ammunition												
M1	9	198										
M109	221	442										
M993	134	167										
M106	91	91										
Tow	68	68										
Other	199	199										
	-------	-------										
Total Ammunition	722	1,165										
Spares	34	56										
Major End Items	77	128										
Other	103	103										
	-------	-------										
CNSM Grand Total	1,541	3,269										
10-Day Grand Sum of Division							101,491	189,790	1,072,088	1,378,869		
Ware							55,273	91,711	1,072,088	1,378,869		
Consumables							46,217	98,079				

TABLE III-6. Typical Daily Consumption (Armored Division)*

	Short Tons	
Class III (Pol)		
M1	721 (40 percent)	
Other tracks	603	
Other ground vehicles	178	
Aviation	173	
Other	142	
Total		1,817
Class V (Ammo)		
M1	230 (17 percent of ammo)	
Tube artillery	513	
MLRS	189	
4.2-in mortar	108	
TOWs	79	
Miscellaneous	233	
Total		1,352
Class VII (Major items)		128
Class IV (Construction)		72
Class IX (Repair parts)		56
TOTAL		**3,425**

* TOE 87000J430 (6 × 4 × 2), first day moderate attack

or survivability) have a mobility and support advantage. If trailers are available commercially, as in the U.S., lighter tracked vehicles can more readily take advantage of them. If roads are bad, including bridging, lighter loads mean fewer detours, restrictions, or offloading of trailers to pass over a weak bridge. Furthermore, as discussed above, a lighter unit requires less massive sustainment (largely fuel) and hence, fewer trucks in the supply train behind them. So long as the logistics are able to keep up with the fighting units, the burden of an excessive weight of resupply is merely more work for the supporting units: A war-fighting deficiency is not easy to perceive. However, should the supply train break and fuel or spares fail to keep up with the unit, the consequences range from cessation of an offensive to annihilation. Thus, any action contributing to the minimization of the logistics burden, such as marked improvement of fuel economy, must be viewed positively.

III.2 Operational

The Operational level of war contains the maneuver aspect of closing with the enemy. The premier example in recent history is the "Hail Mary" move of the VII Corps (and attached British and French forces) during the 100-hour ground war phase of Desert Storm. This maneuver involved the encirclement of the Iraqi forces in Kuwait around their right flank through Iraq. This movement was up to 400 km in 24 hours, with minimal interference from enemy forces. It was essentially an extended road march but without a road. Note that the average speed of the force in the Desert Storm example is very slow, far below that capable by the M1s and M2s in that soil and terrain. The speed of the advance was determined by several factors, most notably the mobility of the supply train cross-country. In future operations, the execution of this type of maneuver may be limited by the enemy rather than by terrain or soil (acting again more likely on the support forces rather than on the principal combat forces). Also, the efficacy of such a maneuver hinges in large part on the blindness of the opponent, which, if not achieved, may nullify the advantages of that maneuver. The point is that IF such a move is a reasonable option, it does offer a great advantage, but its execution is dependent upon the mobility of all the required vehicles. A lighter force will have a maneuver advantage, not because any of its vehicles will necessarily speed over the terrain any faster, but because they will be lessening their dependence on the encumbering supply train.

III.3 Tactical

At the tactical level (the movements of individual divisions and below), where combat is actually executed, mobility is defined not only by speed over rough terrain but by speed on grades, movement in very weak soils, obstacle clearance capability, and simple agility (100-meter dash). These capabilities not only contribute to the general virtue of maneuver but also to mass, and in the event of hostile return fire, to survivability by permitting hasty retreat from incoming indirect fires and by minimizing targetting time for hostile direct fires. For smaller vehicles expected to carry a main tank gun and utilize it appropriately (i.e., to fire on the move with one shot/one kill accuracy), targetting and gun stability demands will dictate extraordinary suspension performance as the weight and track length of the full-sized tank will not be there. Studies have shown (see Fig. III-1) that some form of active suspension will be necessary to achieve a sufficiently stable platform. This may even be necessary simply to achieve sufficient crew comfort to allow a two-man crew to operate effectively for the durations of maneuver and combat called for in the scenarios of future conflict.

At this level, the smaller and lighter vehicle faces a double-edged sword. The lightness contributes favorably for fuel economy, which may be exploited by the designer for either longer time between refueling (a tactical advantage) or a smaller fuel tank (making the vehicle marginally smaller, yet lighter). A smaller vehicle has the advantage of being less readily detectable in various wavelengths, but in particular in the visible. It is also a smaller target, especially if it is lower to the ground in the case of direct fire. For indirect fires, the probability of a hit is in some fashion inversely proportional to the vehicle's footprint, although the quality of the incoming munition may be sufficiently good that the vehicle is perceived as a barn. Smallness would generally be expected to be advantageous, nonetheless.

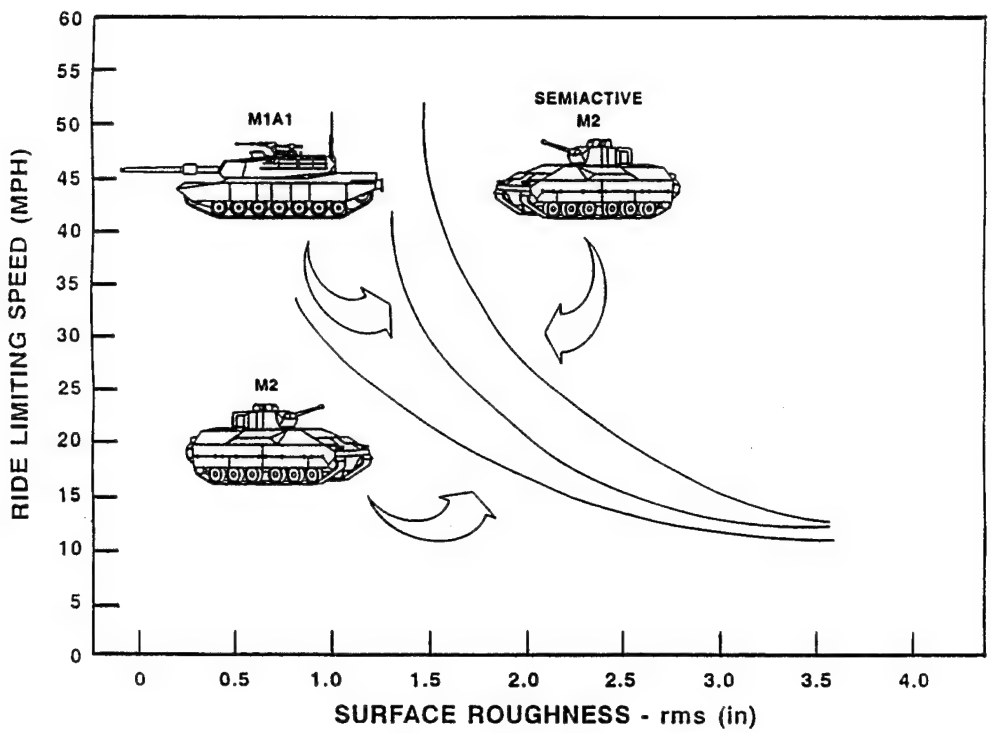


Figure III-1. Simulated ride limited performance

The great disadvantage of the lighter (not necessarily smaller) vehicle is the perception that survivability through armor protection is severely compromised. In fact, the lighter vehicles today (M113A3 through the M2 or M109) do indeed have less armor capability than the M1 group. This was always intended, for these other vehicles are not tank killers and are usually thought of as targets unworthy for a tank to give away its position by firing at it. The M109, an artillery piece, is not even supposed to be within target range of a tank and hence, it is endangered only by indirect fire or by aircraft. Weaker armor thus seems justified. However, if one is to consider a light tank, the situation changes. Such a tank, being a tank killer, will attract tank fire, and being inadequately armored (as with M2 type armor), it will be readily defeated by an opposing tank who first gets off a good shot. This is precisely the situation for the Armored Gun System (AGS). The risk is taken out of the necessity for a light unit (82nd AB) to have some tank-like firepower without the burden of the weight. Such a risk may not make sense across the board for all armored forces.

The opportunity for the future is to achieve tank-like survivability with a lightweight tank, employing protection schemes beyond the smaller target size and smaller EM signature (including visible) that it offers. These approaches include active stealth technology and active (probably electromagnetic) armor, to supplement the light metal alloy and ceramic/composite armor. A small and well-packaged crew may give additional protection to life, if not to the vehicle. Advanced electronic sensing, and in the most optimistic case, an electrothermal cannon (ETC), will allow threat ID, targeting, and destruction at a greater range, thus further protecting the light tank. (This latter speculation implies a desert or arctic-like scenario wherein there is the geographical possibility of long range "vision" and shooting. However, it also puts a further demand on the power consuming suspension system to provide an adequately stable firing platform to succeed at such maximum ranges.) With or without the ETC, a tank-type gun on

such small vehicles implies an unbalanced gun tube. In order to stabilize an unbalanced gun for long range targets, further power consumption is needed in the form of an "active suspension" of the gun itself.

The conclusion at the tactical level is that the smaller, lighter vehicle postulated by the deployability scenarios will have to have supplemental defensive, and possibly, offensive features in order to have an adequate chance of survival in battle. A number of these features will require considerable power ("hotel loads") over and above that required for basic movement (see TABLE III-7). This is what will result in a dramatic impact on the propulsion system. As many of these loads are only loosely related to the vehicle weight, this "hotel" burden will become proportionally more severe for smaller vehicles, especially with a type of electric main gun.

TABLE III-7. Future Electric Power Requirements

• Electric gun (ETC) – 120 mm – 8 rounds stored	600 to 800 hp
• Active suspension	100 to 150 hp
• Fans and accessories	180 hp
• NBC* system	50 to 100 hp
TOTALS	
Without electric gun	**430 hp**
With electric gun	**1,200 hp**

* Nuclear, biological, and chemical

The Implications of Smaller, Lighter, More Powerful Vehicles on Propulsion System Requirements

TABLE III-7 and Figs. III-2 and III-3 show that the space and weight burdens of light, small main combat vehicles (i.e., tanks, etc.) will be excessive unless we can climb above the lines in Figs. III-2 and III-3 by whatever means. Figure III-4, if redone for medium and light vehicles, will show that the diesel engine is also a major chunk of the weight problem. Getting the volume down, especially with the hotel loads being predominately electrical, requires first, addressing electric drive schemes, and secondly, the engine.

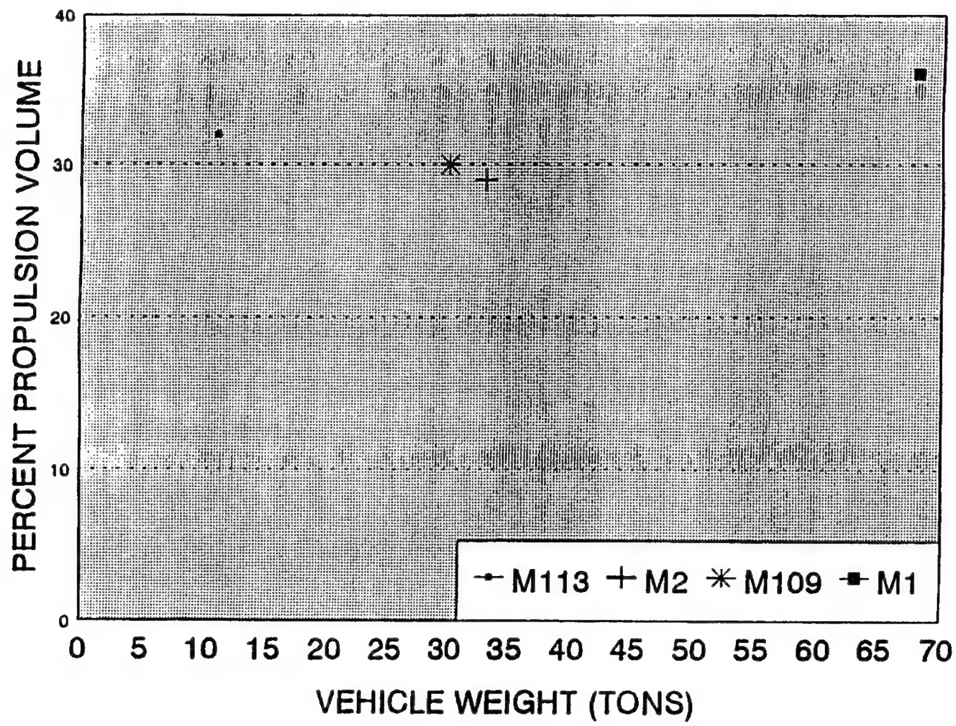


Figure III-2. Percent propulsion volume vs. weight

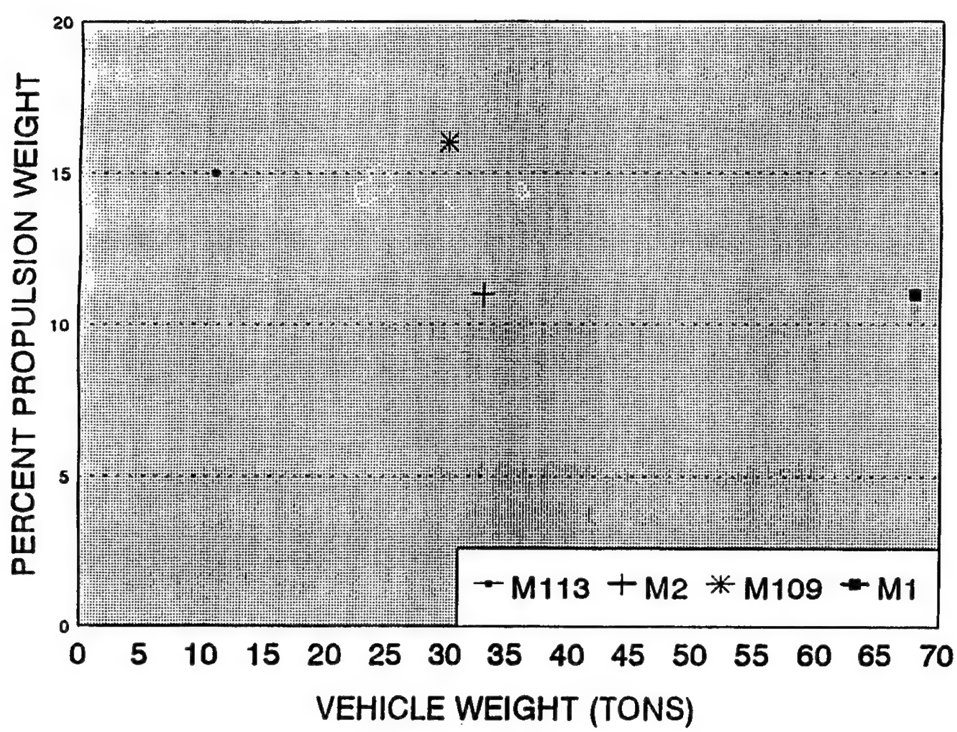


Figure III-3. Percent of propulsion weight vs. total weight

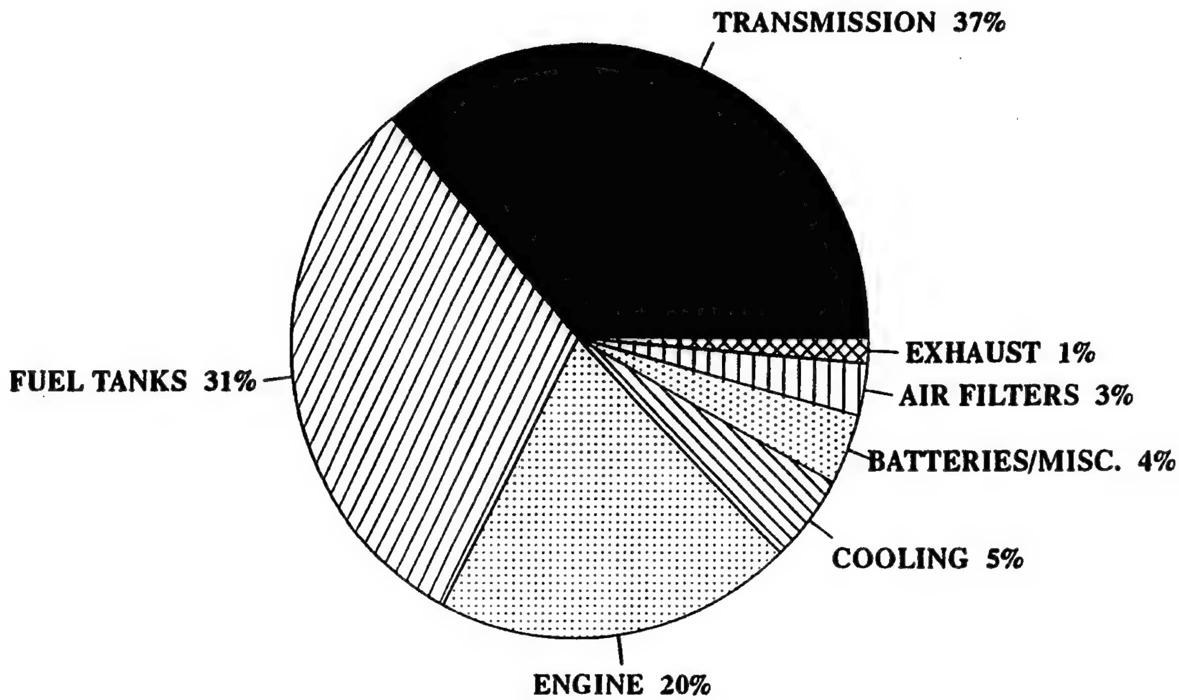


Figure III-4. **M1 powerpack component weight contribution**

APPENDIX V

Performance Requirements Peculiar to Combat Vehicle Propulsion Systems

(Note: Appendix IV is not used.)

APPENDIX V

PERFORMANCE REQUIREMENTS PECULIAR TO COMBAT VEHICLE PROPULSION SYSTEMS

V.1 Propulsion System

The propulsion system includes engine; transmission, including steering and braking; air cleaner; final drives; cooling system, including radiators, fans, plumbing and ducting for engine, hydraulics, electrical, transmission and fuel; exhaust and air inlet/outlet ducting; controls; diagnostic/prognostic/maintainability components; design features and software; infrared, noise, and smoke signature suppression; vehicle hydraulic, electrical and pneumatic (if any) power generation equipment and conditioning as required for propulsion; batteries; fuel cells and fuel fill and delivery systems; nuclear, biological, and chemical (NBC) interfaces; fire suppression volume as required; propulsion system mounting; other propulsion system ancillary equipment.

V.2 Performance Requirements

The performance requirements and environmental conditions under which combat vehicle propulsion systems must operate are typically much more severe than those of commercial propulsion systems. Some of the major differences are discussed in this section.

V.2.1 Duty cycle

Combat vehicles require high peak power to meet mobility and other performance requirements (e.g., speed on grade and vehicle acceleration). The combat vehicle load cycle is, however, heavily weighted toward light load conditions, including a high percentage of operating time at engine idle for both 24-hour battlefield day (BFD) and peacetime annual (PA) duty cycles, as illustrated in TABLES V-1 and V-2.

It is obvious that the above load cycles are heavily weighted toward low-load conditions. This is in contrast to heavy-duty, over-the-road truck operation where, except possibly for long wintertime idling, much of the time is spent at three-quarters to full power. It follows then, that high-power fuel consumption is not as important for combat vehicle propulsion systems as for heavy-duty truck propulsion systems. As a result, techniques such as turbocompounding, which improve primarily high-load fuel consumption and may even adversely affect low-load fuel consumption, may not increase propulsion system compactness for combat vehicles. There are other differences. For example, since combat vehicles operate very little time at full-load/power, heavy load durability may not be as important for combat vehicle propulsion systems as for heavy-duty truck propulsion systems, while short-term reliability may be more important.

TABLE V-1. Main Battle Tank 24-Hour Battlefield Day Mission Definition

(For fuel consumption calculations)

Operating Condition/ Net Power Outputs	Duration, hr	Vehicle Speed, mph/kph	Rolling Resistance, lb/ton / kg/tonne
Engine Starts, (6×) From 400°F engine soak	(TBD for engine)	0	N/A
Electric Power 5 kW from alternator Max. NBC load if ON	0.500	0	N/A
Idle 5 kW from alternator Max. NBC load if ON	2.330	0	N/A
Tactical engine idle 5 kW auxiliary load Max. NBC load if ON	0.917		
Secondary Road Level road load 5 kW auxiliary load Max. NBC load if ON	3.400	24.9/40	125/62.5
Cross Country Level road load 5 kW auxiliary load Max. NBC load if ON	3.333	16.8/27	250/125
Silent Watch 5 kW from alternator Max. NBC load if ON	1.500	0	N/A
OFF All power requirements = 0	12.000	0	N/A
TOTAL	**24.000**		

Input Data Required	Output Data Required*	Standard Conditions
System definition GVW Deviations from standard conditions NBC ON or OFF Data source and date	Sprocket hp Engine output hp Engine output speed Installed (inlet & exhaust losses) BSFC mpg Fuel quantity used (gal.) Fuel flow rate (pph & gph)	Ambient temp. 87°F (30°C) Altitude = 500 ft (150 m) Fuel LHV = 18,500 Btu/lb (43,500 kJ/kg) Density = 7.05 lb/gal. (0.847 kg/L) Alternator/aux. load conversion eff. @ 5 kW = 80 percent

* English units only

TABLE V-2. Main Battle Tank Peacetime Annual Use Definition

(For fuel consumption calculations)

Operating Condition/ Net Power Outputs	Duration, hr	Vehicle Speed, mph/kph	Rolling Resistance, lb/ton / kg/tonne
Engine Starts, (120×)	(TBD for engine)	0	N/A
From 87°F engine soak			
Electric Power	33.0	0	N/A
5 kW from alternator			
Max. NBC load if ON			
Idle		0	N/A
5 kW from alternator	125.0		
Max. NBC load if ON			
Tactical engine idle			
5 kW auxiliary load	50.0		
Max. NBC load if ON			
Secondary Road			
Level road load	12.0	5.0/8	125/62.5
5 kW auxiliary load	14.9	24.9/40	125/62.5
Max. NBC load if ON	9.0	34.8/56	125/62.5
Cross Country	11.3	16.8/27	250/125
Level road load			
5 kW auxiliary load			
Max. NBC load if ON			
TOTAL	**255.2 + Starts**		

Input Data Required	Output Data Required*	Standard Conditions
System definition	Sprocket hp	Ambient temp. 87°F (30°C)
GVW	Engine output hp	Altitude = 500 ft (150 m)
Deviations from standard conditions	Engine output speed	Fuel LHV = 18,500 Btu/lb (43,500 kJ/kg)
NBC ON or OFF	Installed (inlet & exhaust losses) BSFC	Density = 7.05 lb/gal. (0.847 kg/L)
Data source and date	Fuel quantity used (gal.)	Alternator/auxiliary load conversion efficiency @ 5 kW = 80 percent
	Fuel flow rate (pph & gph)	
	mpg	

* English units only

V.2.1.1 Altitude and temperature power degradation

The available power at the sprocket at ambient conditions defined by NATO Allied Engineering Publication (AEP-5) at 150-m altitude will not degrade in excess of 15 percent at 49°C and 20 percent at 20°C, 1,800-m altitude. These requirements reflect the reality that power and cooling system capacities are naturally degraded by less dense atmospheric conditions. Yet, it also limits that degradation so that vehicle performance capabilities are not so severely attenuated that combat worthiness is compromised.

V.2.2 Cooling capability

The power pack shall be capable of providing continuous cooling with a 30 percent safety margin. The required cooling shall be provided at all ambient conditions between –51°C to +49°C at 750 mm Hg, including 0.7 tractive effort (TE)/gross vehicle weight (GVW), all braking conditions, and gear-engaged engine idle cooling at environmental extremes. Cooling the propulsion system in a combat vehicle is especially challenging because it is enclosed in armor. Also, the close fit between the propulsion system components and the surrounding armored engine compartment permits very little convective and radiation heat transfer to take place.

V.2.3 Incline

Combat vehicle propulsion systems are required to operate continuously at 60 percent fore and aft and 40 percent side-to-side inclination, and withstand the lateral accelerations of high-speed turns to the maximum capability of the vehicle at a slip of 0.7 TE/GVW.

V.2.4 Fuel tolerance

V.2.4.1 Primary fuels

The primary fuels are JP-8 (MIL-T-83133) in all ambient temperatures and the fuels of federal specification VV-F-800 (DF-A, DF-1, DF-2) at the appropriate ambient temperatures.

V.2.4.2 Alternate fuels

Combat propulsion systems must also be capable of operating on MIL-F-53080 (Type I and Type II fuels) at normal ambient conditions as defined by NATO AEP-5 at 150-m altitude.

V.2.4.3 Emergency fuels

The propulsion system must also be capable of limited operation on gasoline (MIL-F-46015) with some degraded performance and durability allowed.

V.2.5 Starting capability

The following defines the ambient and propulsion system hardware temperatures for combat engine startability using primary fuels:

V.2.5.1 <u>–10°C and above</u>

Start in 1 minute or less using DF-2 and JP-8 fuels independently, without external aids or kits.

V.2.5.2 <u>–30°C to –10°C</u>

Start in 1 minute or less using DF-1 and JP-8 fuels independently, without external aids or kits.

V.2.5.3 <u>–50°C to –30°C</u>

Start in two minutes or less from initiation of cranking using DF-A and JP-8 fuels independently; external aids or kits are permitted. The engine must be started within 30 minutes from initiation of the arctic kit-aided start sequence.

V.2.6 <u>Fording</u>

The combat propulsion system must be capable of wet compartment fording to include shutdown and restart while immersed underwater in an unprepared mode at a depth of 1.5 m.

V.2.7 <u>Shock and vibration</u>

High speed, cross-country operations produce high shock and vibration loadings and lots of dust. Severe shock and vibration impulses are imposed on combat propulsion systems from basic vehicle operation including cross-country and other terrains (30 g), gun firing (60 g), and ballistic conditions (55 g), all axes-maximum.

V.2.8 <u>Air cleaner system</u>

Combat vehicles typically operate in extremely heavy dust environments, which is a serious problem compared to commercial trucks. The air cleaner system in a combat vehicle must be capable of operating for a minimum of 200 service-free hours with repetitions of 10-minute intervals under conditions of 20, 40, 60, 80, and 100 percent sequentially, of maximum rated airflow and zero visibility dust.

V.2.9 <u>Exhaust emissions</u>

Even though combat vehicles/engines are not required to meet the Environmental Protection Agency standards for exhaust emissions, such emissions must be reduced to minimize detectability by an enemy. Combat engines shall not exceed 8 percent opacity or equivalent under all steady-state conditions and 12 percent opacity or equivalent under all transient conditions.

V.2.10 <u>Reliability and durability</u>

Combat propulsion system reliability and durability requirements differ drastically from commercial propulsion systems. To achieve the required compactness in a combat propulsion system, the specific output (horsepower per cubic inch) is typically higher for combat engines.

Reliability is crucially important for propulsion systems in combat vehicles; you cannot ask for a postponement in either the shooting or the war while you repair your propulsion system! Inoperability at a critical moment may be fatal rather than inconvenient. However, during combat, the lifetime of a combat vehicle is measured in the tens to hundreds of miles. There is a big difference in propulsion system/engine life expectancy of combat versus commercial propulsion systems. A modern combat propulsion system design objective is to operate for up to 10,000 miles before overhaul, whereas commercial trucks are running well over 250,000 miles before overhaul. In peacetime, combat vehicles typically operate less than 1,000 miles per year, whereas commercial trucks typically run up to 250,000 miles annually. This large disparity in actual propulsion system/engine life is a reflection of the very severe operating and environmental conditions under which combat vehicles must operate.

Upon completion of advanced development, the propulsion system is required to undergo 1,000 hours of endurance testing, including a 400-hour NATO AEP-5 test and a 600-hour TACOM Tank Mission Profile Test.

V.3 Extremes (Environmental, Terrain, Fuels, etc.)

Since the location of future hostilities is not known, present Army policy is that all combat vehicles must be able (in some cases using add-on kits) to operate with minimal performance degradation at the environmental extremes found on the planet Earth. The specified temperature extremes (–60 to 120°F, for example) are considerably greater than those experienced by over-the-road, heavy-duty trucks. The same comment is true of the terrain, which ranges from cross country to secondary roads, snow to ice to sand to mud for combat vehicles. While the Army does have specified fuels, the practicalities of the battlefield essentially guarantee that the engine will need to be operable, even if not at full performance, on a variety of fuels.

V.4 Armor Enclosure

In combat vehicles, the entire propulsion system is enclosed by armor with very little air circulation. This in itself adversely affects cooling. The protection-dictated locations of inlet grilles means that, for cooling, little or no advantage can be taken from the forward motion of the vehicle. Also, typically, full power is used under grade or heavy traction conditions where vehicle speeds are low. Furthermore, the necessarily large flows of combustion and cooling air must come in and out through restrictive ballistic grilles. Because of restricted air flow through the ballistic grilles, as much as a 10 to 15 percent penalty in engine power may result, which is two to three times the power penalty for commercial, heavy-duty trucks. It follows, then, that a propulsion system and its components optimized for commercial, heavy-duty trucks will not be equally optimum for combat vehicles.

V.5 Component Trade-offs

The need for coverage by armor of the propulsion system and the resulting compactness requirement means that trade-offs between components are different between commercial and combat vehicles. For example, in a commercial vehicle, the preferred solution to a cooling problem may be a slightly larger radiator. On the other hand, in a combat vehicle with all of the implications of enlarging the armored volume, the preferred solution may be to operate the engine

and its coolant system at a higher temperature to increase the rate of heat transfer. The point is that the research and development required is markedly different between the larger radiator and the higher engine temperature approach. Consequently, it cannot be assumed that the research and development needed to produce compact propulsion systems for combat vehicles will result from the ever on-going commercial research and development. Therefore, the technology advances required to meet future Army-unique requirements can only be attained through Army-funded research and development programs.

APPENDIX VI

Optimizing Combat Vehicle Propulsion Systems

APPENDIX VI

OPTIMIZING COMBAT VEHICLE PROPULSION SYSTEMS

VI.1 Comments

Volume compactness can be improved either of two ways: improving the integration of components to produce a more volume compact system or improving the volume compactness of individual components to form a more volume compact system. Weight compactness is generally achieved either by a system design change or by increased material stresses. However, regardless of whether one is considering weight or volume compactness, optimizing a multicomponent system is a complex exercise. Let us, therefore, comment first on the optimization process and then on the unique performance requirements for propulsion systems for combat vehicles.

Large sums of money, well beyond those available to the Army, have been and continue to be invested by industry in optimizing propulsion systems for use in commercial, heavy-duty applications of intermittent combustion engines. However, these monies have largely concentrated on increasing reliability and durability, decreased fuel consumption, and meeting emission regulations. Minimal attention has been paid to achieving HPD propulsion systems.

Since HPD is relatively unimportant for tactical vehicles, the Army has been able to "buy commercial," i.e., adapt propulsion systems developed using primarily commercial funds. As was shown in Section 3, Volume I, HPD propulsion systems are absolutely essential for Army combat vehicles.

In any case, the components and configuration needed for optimum performance of any system depends upon the performance requirements to be met by that system. As is discussed in Section 5, Volume I, the performance requirements for propulsion systems for combat vehicles differ significantly from the propulsion system performance requirements for commercial heavy-duty vehicles.

Optimizing any multicomponent system is difficult and is best approached through the use of as-accurate-as-possible analytical models. Usefully accurate but complex analytical models of the engine component of the propulsion system have been developed [initially under the U.S. Army Tank-Automotive and Armaments Command (TACOM) sponsorship] and are used by almost all commercial developers of engines. In addition, some vehicle manufacturers have developed more total system-oriented models that include other related components of the propulsion system. TACOM has simplified spreadsheet analysis capabilities, but no analytical models have been developed that are useful for analysis of combat vehicle propulsion systems. As a result, when currently changing the characteristics of one component of the propulsion system for combat vehicles, spreadsheet analysis plus intuition and past experience must be used to judge the interaction between components and the effects of changing one component on other components and consequently, on the compactness of the entire system. The Army needs access to a propulsion system model in order to minimize the time and money required to develop extremely compact propulsion systems for combat vehicles.

APPENDIX VII

The Baseline for Judging Improvements in Propulsion System Compactness

APPENDIX VII

THE BASELINE FOR JUDGING IMPROVEMENTS IN PROPULSION SYSTEM COMPACTNESS

VII.1 dAIPS Engine Baseline Description

The dAIPS is a propulsion system powered by a diesel engine which underwent partial development for the next generation of heavy armored combat vehicles. The integrated system includes a high horsepower, advanced technology diesel engine; a compact, state-of-the-art, cross-drive, hydrokinetic transmission with integral brakes and hydrostatic steering; high effectiveness, compact oil-to-air cooling; self-cleaning air filtration; and electrical, hydraulic, and air auxiliary power. The installation layout of the entire propulsion for a rear drive tank application is depicted in Fig. VII-1. A picture of the entire propulsion system is shown in Fig. VII-2. The diesel engine for this system is the technical baseline case for the BRC study from which propulsion system power density improvements via recommended research are to be made. Installation volumes attributable to the engine and the rest of the propulsion system componentry are provided in TABLE 9-1 of Appendix 9, Volume I. A more detailed description of the engine will be presented here.

The AIPS diesel engine is a four-stroke cycle, direct injected, turbocharged, aftercooled, low-heat rejection engine with strategic oil cooling. The V-12 engine has a displacement of 1,682 cu. in. and has a 1,450 bhp rating at 2,600 rpm. An illustration of the engine alone is provided in Fig. VII-3. Additional engine characteristics, including dimensions, are provided in TABLE VII-1. Engine power and torque performance curves are provided in Fig. VII-4, and a fuel map is shown in Fig. VII-5. It should be noted that these performance curves are preliminary projections and have not been demonstrated. Demonstrated numbers are given in TABLE VII-1. A cross section of the engine is illustrated in Fig. VII-6.

VII.1.1 Fuel injection system

An Electronic Controlled Injection (ECI) system is employed on the dAIPS engine. The ECI system has one unit injector per cylinder. Each injector has one solenoid valve controlling the timing and metering of the fuel. Peak injection pressures on the order of 23,000 psi are generated by the engine's overhead cam and a hydraulic link pumping arrangement within each injector. Fuel is supplied to each injector via a mechanical pump with integral regulator at a common rail pressure of 150 psi. Maximum fuel flow of the injector is 400 cu. mm/stroke, and minimum flow is 25 cu. mm/stroke. Maximum injection duration at rated power is 35 crank angle degrees. A schematic of the ECI injector is shown in Fig. VII-7, and principles of operation are given in Fig. VII-8.

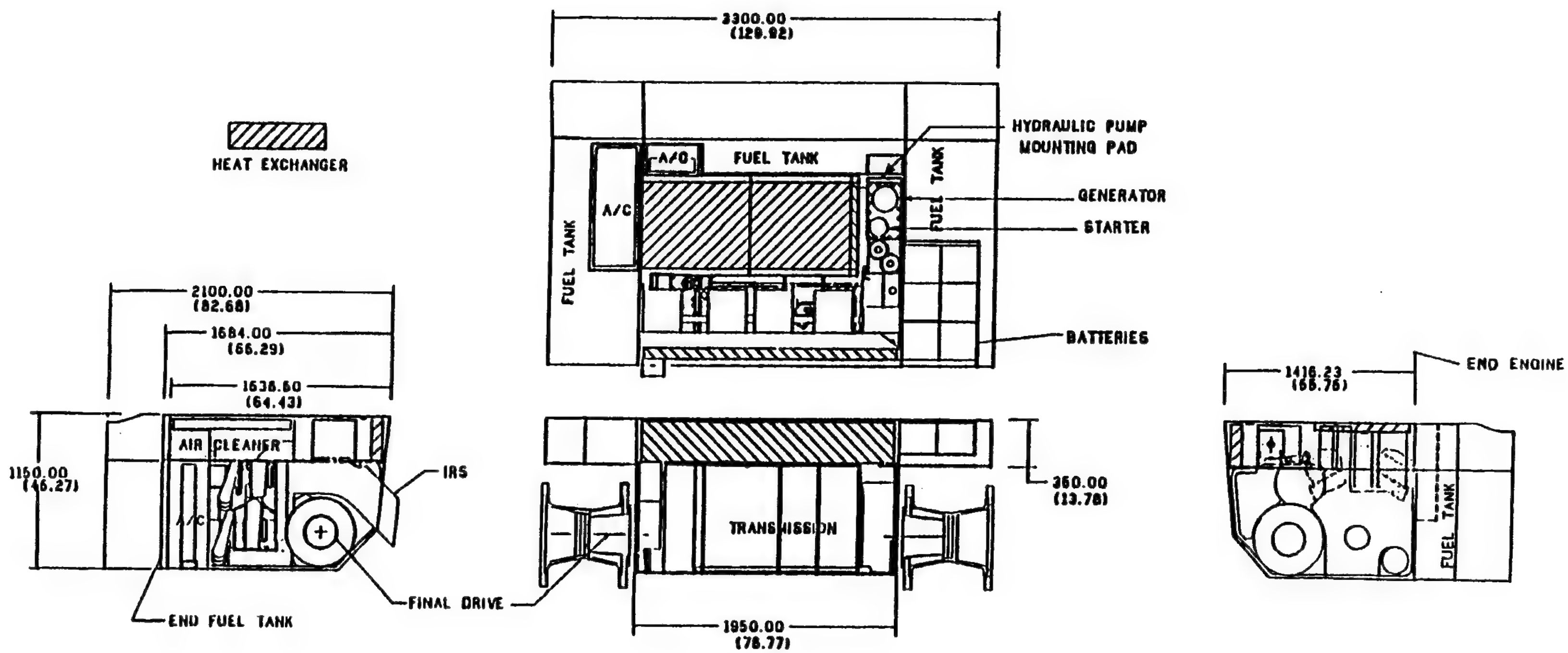


Figure VII-1. dAIPS V-12 installation layout

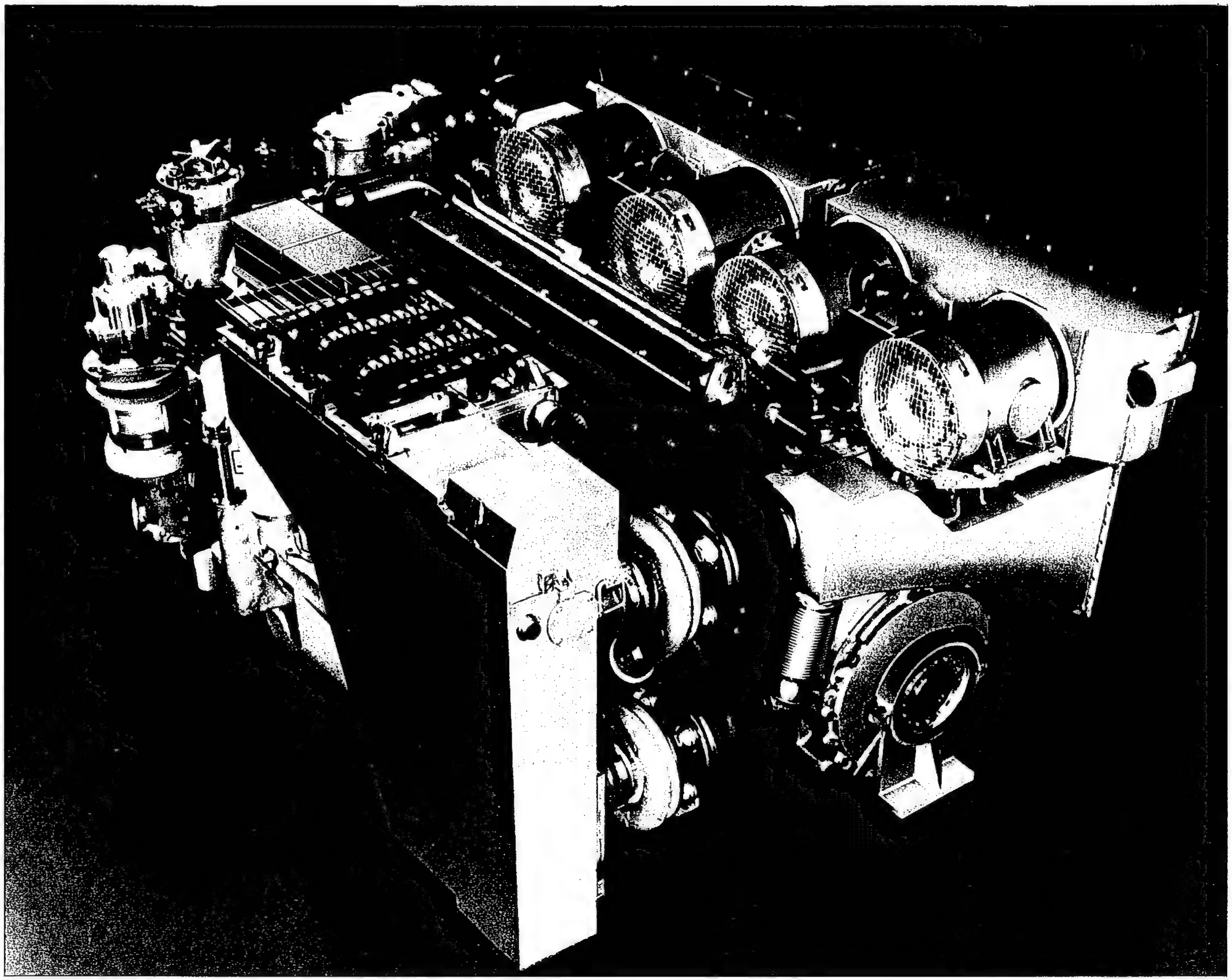


Figure VII-2. <u>dAIPS propulsion system</u>

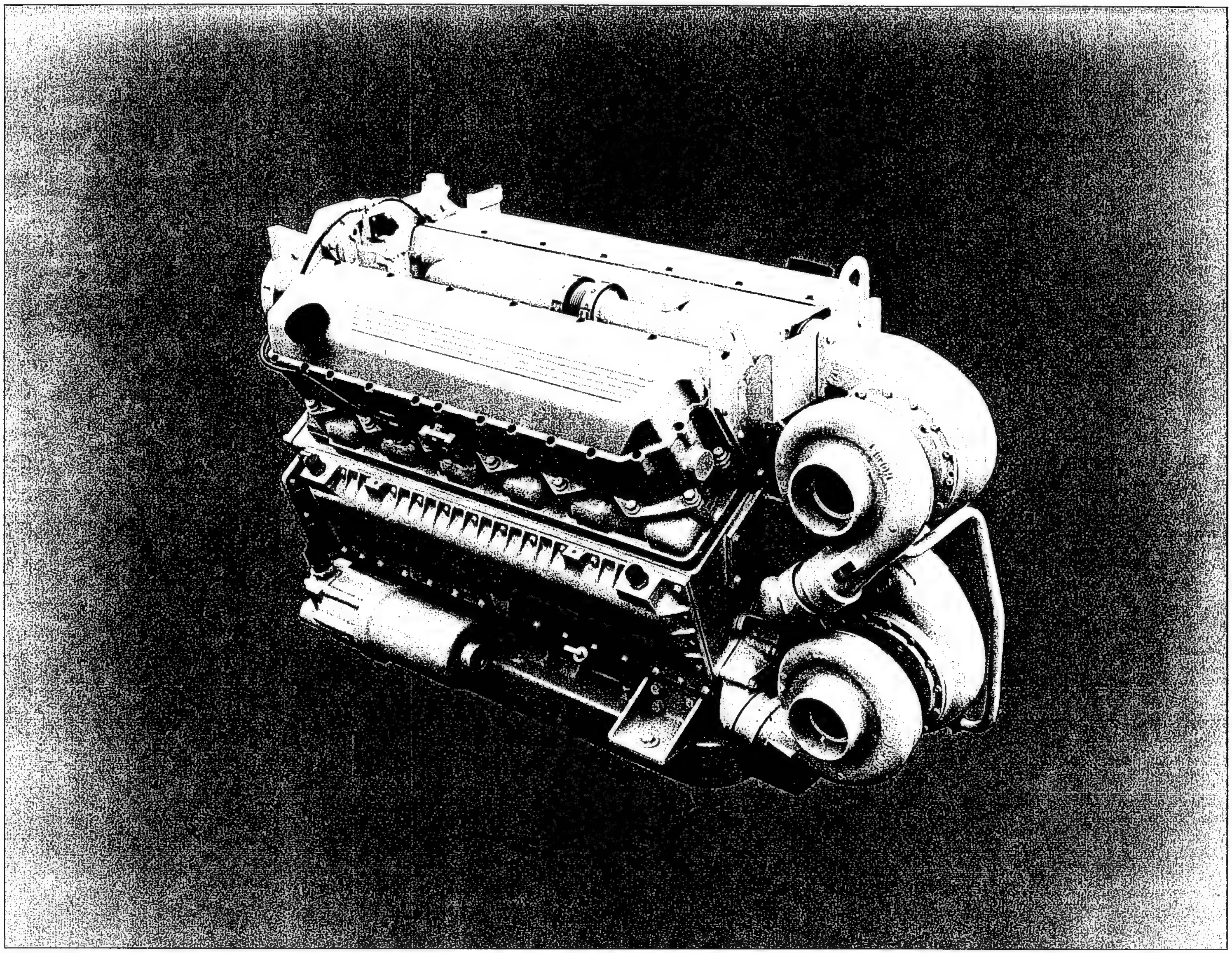


Figure VII-3. dAIPS engine

TABLE VII-1. dAIPS Engine Data Sheet

Number of Cylinders	12
Cylinder Arrangement	60°v
Bore, in	5.906
Stroke, in.	5.118
Displacement, cu. in.	1,682
Power hp at Rated Speed, rpm	(Nominal 1,500) 1,450 installed at 2,600
Max Torque at Speed, rpm	3,500* at 1,560; 2,983** at 1,560
BMEP at Rated, psi	263
BMEP at Torque Peak, psi	315* at 1,560; 268** at 1,560
Mean Piston Speed, ft/min	2,218
Compression Ratio	15.0
Injection Pressure, KSI	23.0
Peak Cylinder Pressure, psi	2,600
Air Consumption, lb/s	3.86
Turbo Pressure Ratio	3.50
Air/Fuel Ratio at Rated	28
BSFC (lb/bhp-hr) at Rated	0.320* (0.371)**
BSFC (lb/bhp-hr) at Torque Peak	0.307* (0.356)**
Cooling Fluid	Oil
Specific Heat Rejection, Btu/hp-min	19.0* (21.6)**
Max Coolant Temperature, °F	340
Cooling Fan Power, hp	120
Weight, lb	4,170
Height, in.	37.4
Width, in.	30.6
Length, in.	68.9
Volume (dunk), cu. ft	34.0

* Target
** Demonstrated

VII.1.2 Cylinder block

The V-12 block is a cast gray iron, one-piece design with a 60° bank angle. The block is configured for a mid-stop liner. Oil rifles are drilled into the block to minimize external lines. The block has integral charge air cooler and intake manifold housings on the outboard side of the liners. Main bearing caps are a four-bolt design.

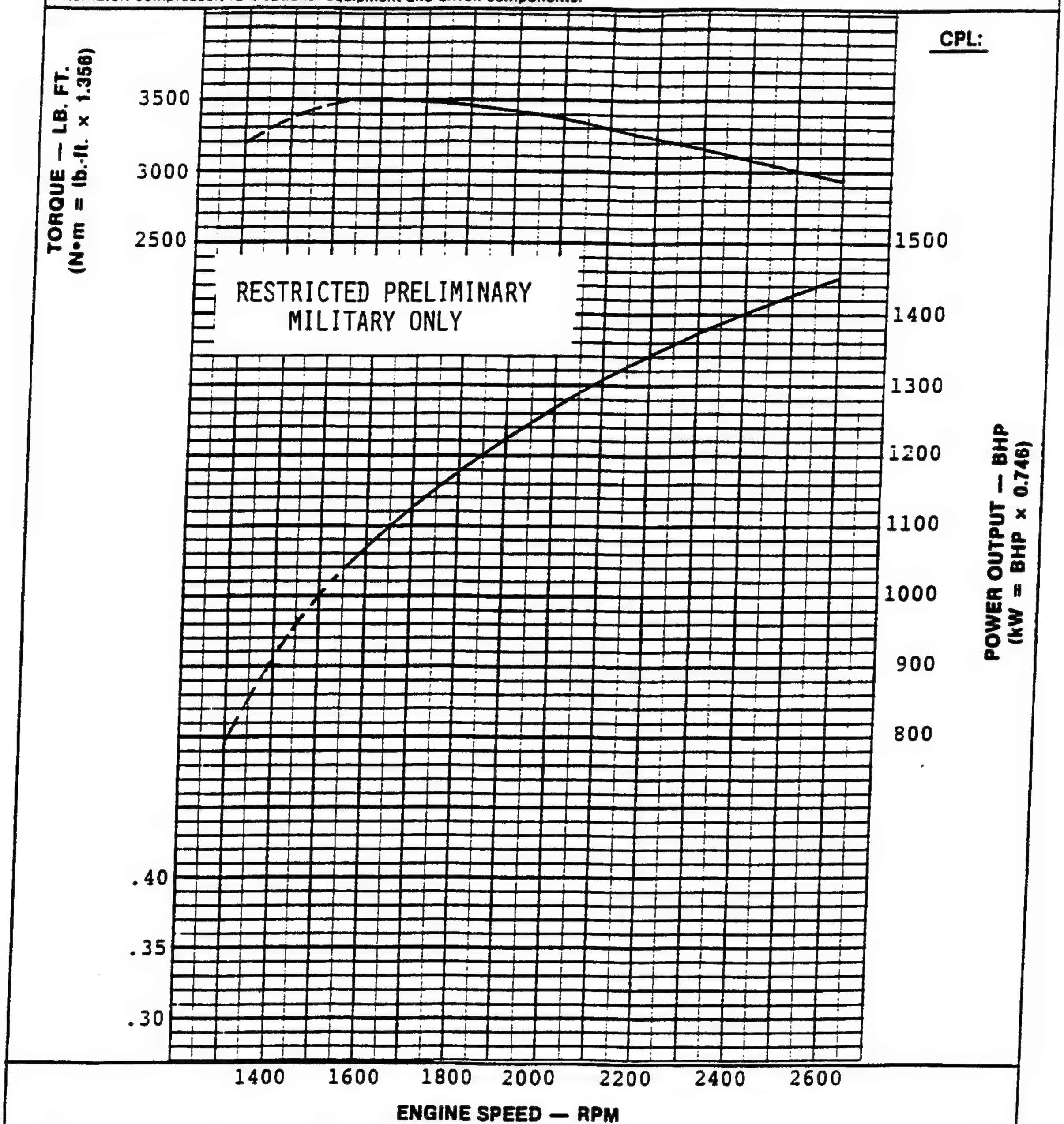
CUMMINS ENGINE COMPANY, INC.
Columbus, Indiana 47201
ENGINE PERFORMANCE CURVE

ENGINE MODEL:	CURVE NUMBER:	
XAV28-1450	RP-4422-02	
ASPIRATION:	**DATE:**	**BY:**
TURBO. & AFTERCOOLED	10/27/86	JFS

DISPLACEMENT: 1682 in³ (27.6 litre) **NO. OF CYLINDERS:** 12
BORE: 5.9 in. (150mm) **STROKE:** 5.12 in. (130mm)

RATING: HP (kW) @ RPM
1450 (1082) 2600

All data is based on the engine operating with fuel system, water pump, lubricating oil pump, air cleaner, and muffler; not included are alternator, compressor, fan, optional equipment and driven components.



Curves shown above represent gross engine performance capabilities obtained and corrected in accordance with SAE J1349 conditions of 29.61 in. Hg (100 kPa) barometric pressure [300 ft. (90 m) altitude], 77°F (25°C) inlet air temperature, and 0.30 in. Hg (1 kPa) water vapor pressure with No. 2 diesel fuel. The engine may be operated without changing the fuel setting up to ft. (m) altitude. For sustained operation at high load factors at higher altitudes, the fuel rate of the engine should be adjusted to limit performance by 4% per 1,000 ft. (300 m) above ft. (m) altitude. The engine altitude capability is based upon an inlet temperature representative of the ambient temperature for that altitude. See reverse side for application rating guidelines.

STANDARDS DEPT. **CERTIFIED WITHIN 5%:** **CHIEF ENGINEER**

Form 4738-1282R

Figure VII-4. <u>dAIPS engine power and torque performance curves</u>

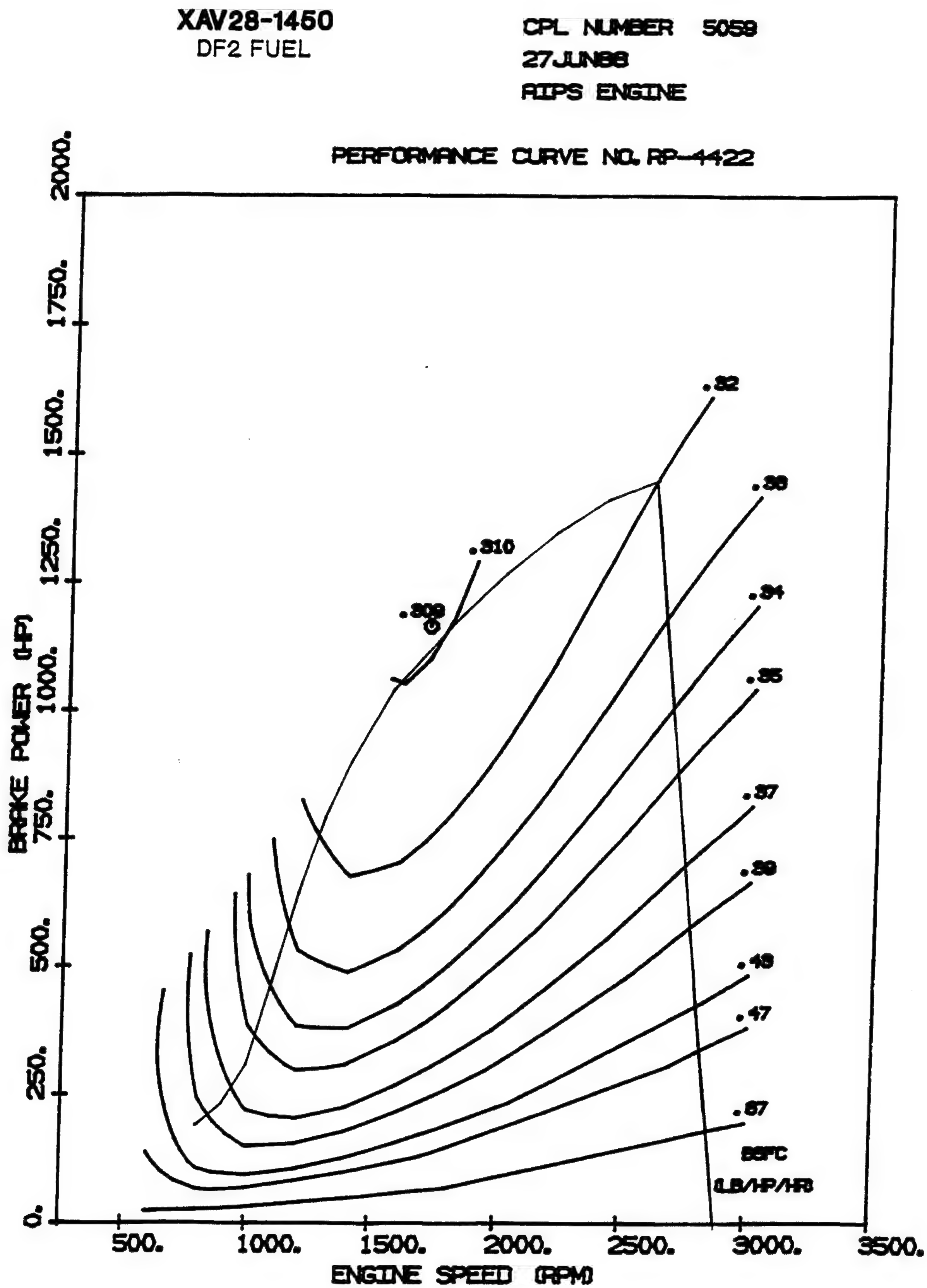


Figure VII-5. <u>dAIPS fuel map</u>

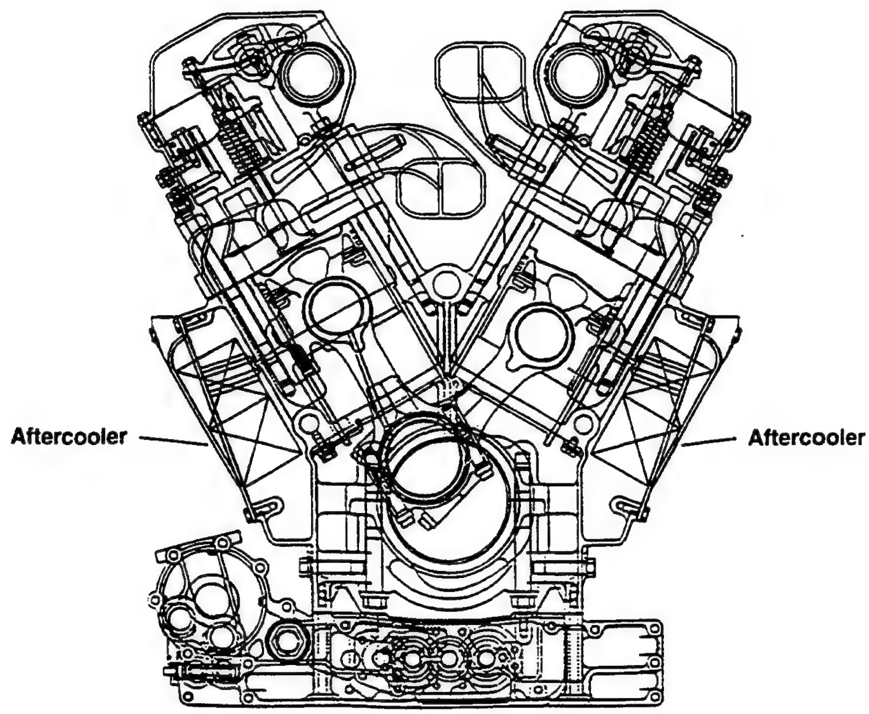


Figure VII-6. V-12 dAIPS engine cross section

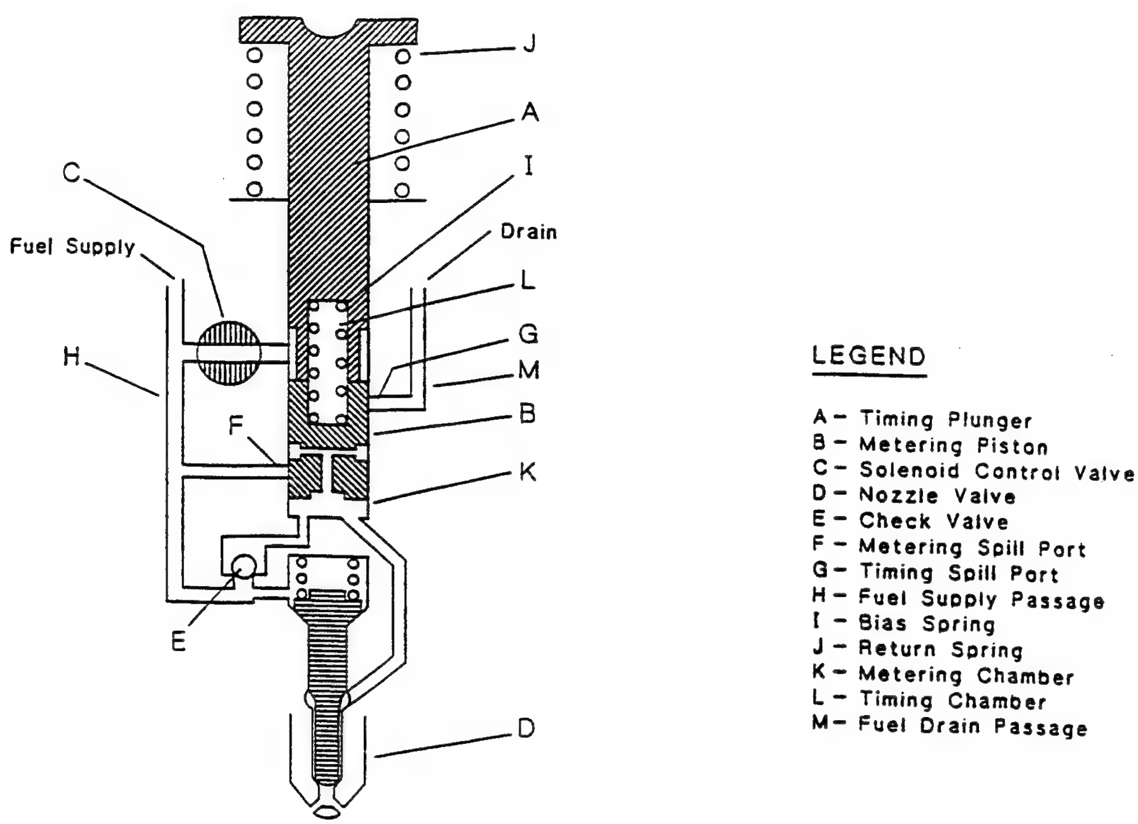


Figure VII-7. ECI injector schematic

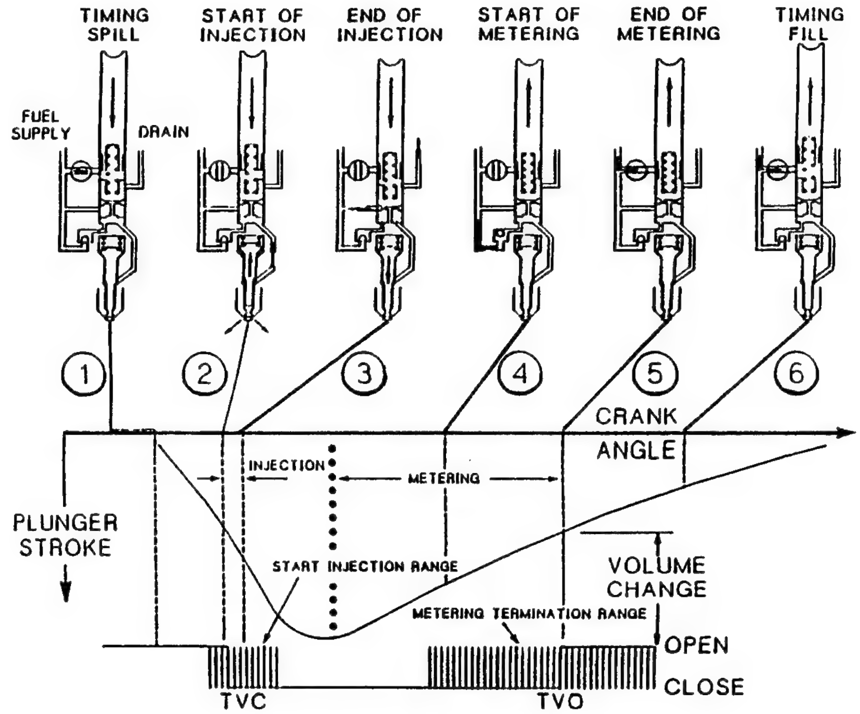


Figure VII-8. **ECI principles of operation**

VII.1.3 Cylinder head

Originally, the cylinder heads were made with cast ductile iron with provisions for strategic oil cooling in the valve bridge areas. All performance running up to this point was done with this head design. Design studies and evaluations on single cylinder engines focused on no oil cooling and inconel materials for future heads. Each head has one intake and one exhaust port with two valves for each port. Exhaust ports are insulated with a stainless steel/ceramic fiber composite lining. The cylinder firedeck is insulated with a thermal barrier coating. A cross section of the cylinder head and the overhead arrangement is given in Fig. VII-9.

VII.1.4 Piston

The piston is an articulated design. The top ring belt region of the piston is made of a high-temperature capable alloy steel. Thermal barrier coatings are employed on the piston dome. Zirconia and mullite coatings are candidates. An aluminum skirt is used for the piston's bottom half. A hollow steel piston pin connects the steel dome to the aluminum skirt. The underside of the dome having a cock-tail shaker design is oil cooled via a single piston oil supply nozzle. The piston's cross section is depicted in Fig. VII-10.

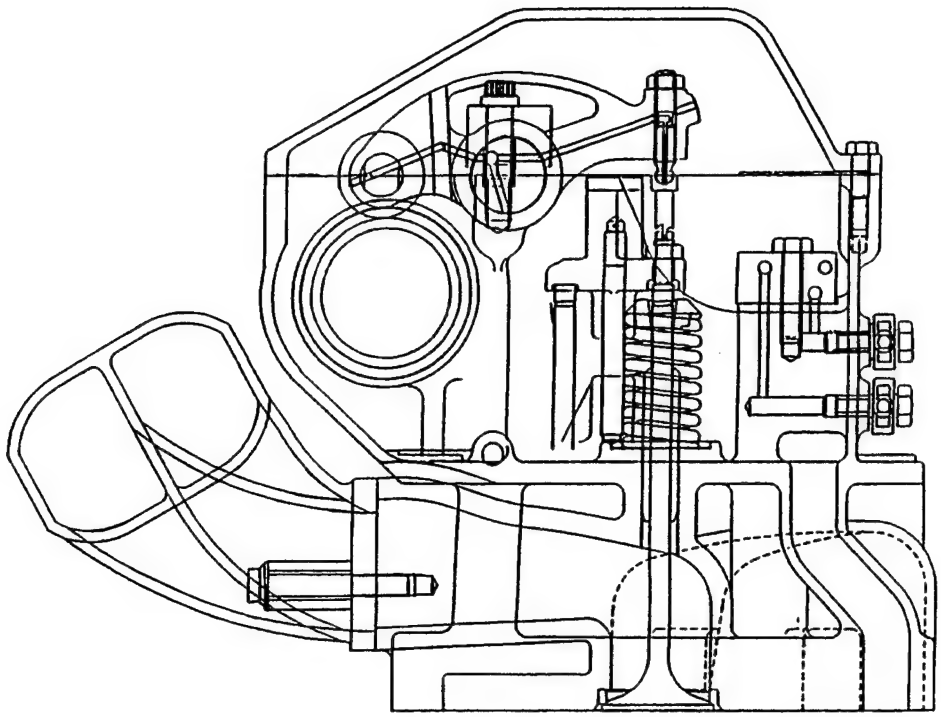


Figure VII-9. <u>Cylinder head cross section</u>

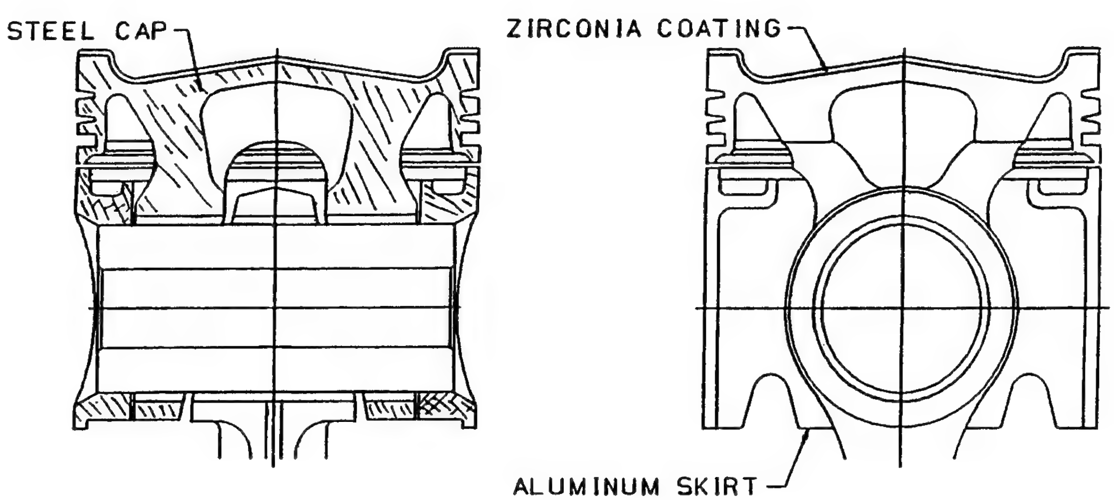


Figure VII-10. <u>Piston cross section</u>

VII.1.5 Liner

The cylinder is made of cast alloyed gray iron. A thin wear coating of chrome oxide is used to enhance wear resistance. Strategic oil cooling is used to cool the liner near the top ring groove region. The liner with the lubricant cooling arrangement is shown in Fig. VII-11.

VII.1.6 Valves

The AIPS diesel engine is a four-valve design. The valves are actuated by the overhead cam system by ductile iron rocker levers. This can be seen in the head drawing of Fig. VII-9. The valve head material is forged Waspalloy, and the stems are made of heat resistant alloy steel. Although the valve seat design was not finalized, the use of monolithic zirconia seats were contemplated to minimize heat transfer through the valve seat area.

VII.1.7 Crankshaft and connecting rods

The crankshaft is made of steel and of a forged in-place design. It has induction hardened fillets and journals. The connecting rods of rectangular cross section shank design are forged alloy steel and are through hardened. The piston and connecting rod assemblies are installed and removed through the cylinder liner bore.

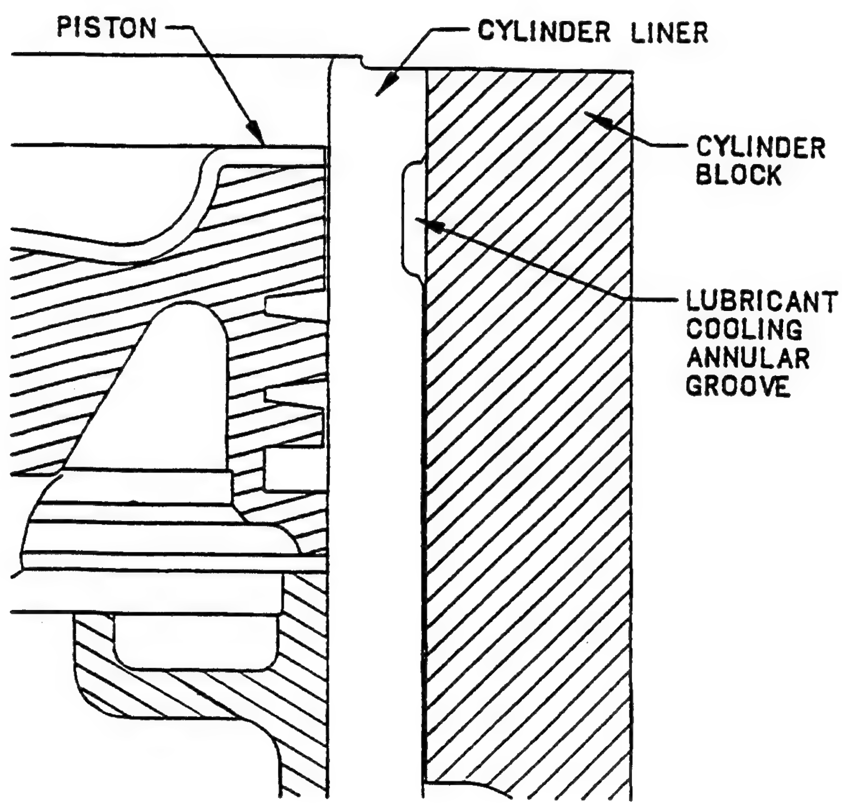


Figure VII-11. **Liner/piston/ring cooling**

VII.1.8 Manifolds

The intake manifolds are located on the outer side of the vee block, while the exhaust manifolds are located on the inner side. See cross section of the engine, Fig. VII-6. The intakes are integral to the block. The exhaust manifolds are made of high-temperature stainless steel with bellows located between sets of three cylinders to provide for expansions and contractions. These run the length of the vee to provide exhaust gas to the turbocharger turbine.

VII.1.9 Aftercoolers

The aftercoolers are located on the outboard side of the engine block, one on each side. See Fig. VII-6 for location. These are plate-fin type, air-to-oil heat exchangers made of stainless steel. Engine oil is used as coolant.

VII.1.10 Turbomachinery

The turbo compressor is a single-stage radial unit that uses map-width enhancement to achieve wide operating range at high pressure ratios. A compressor map showing this wide operating range is shown in Fig. VII-12. The turbocharger turbine is a single radial design that utilizes variable geometry to obtain improved engine transient response. The variable geometry design uses a moving wall approach that controls gas exit area.

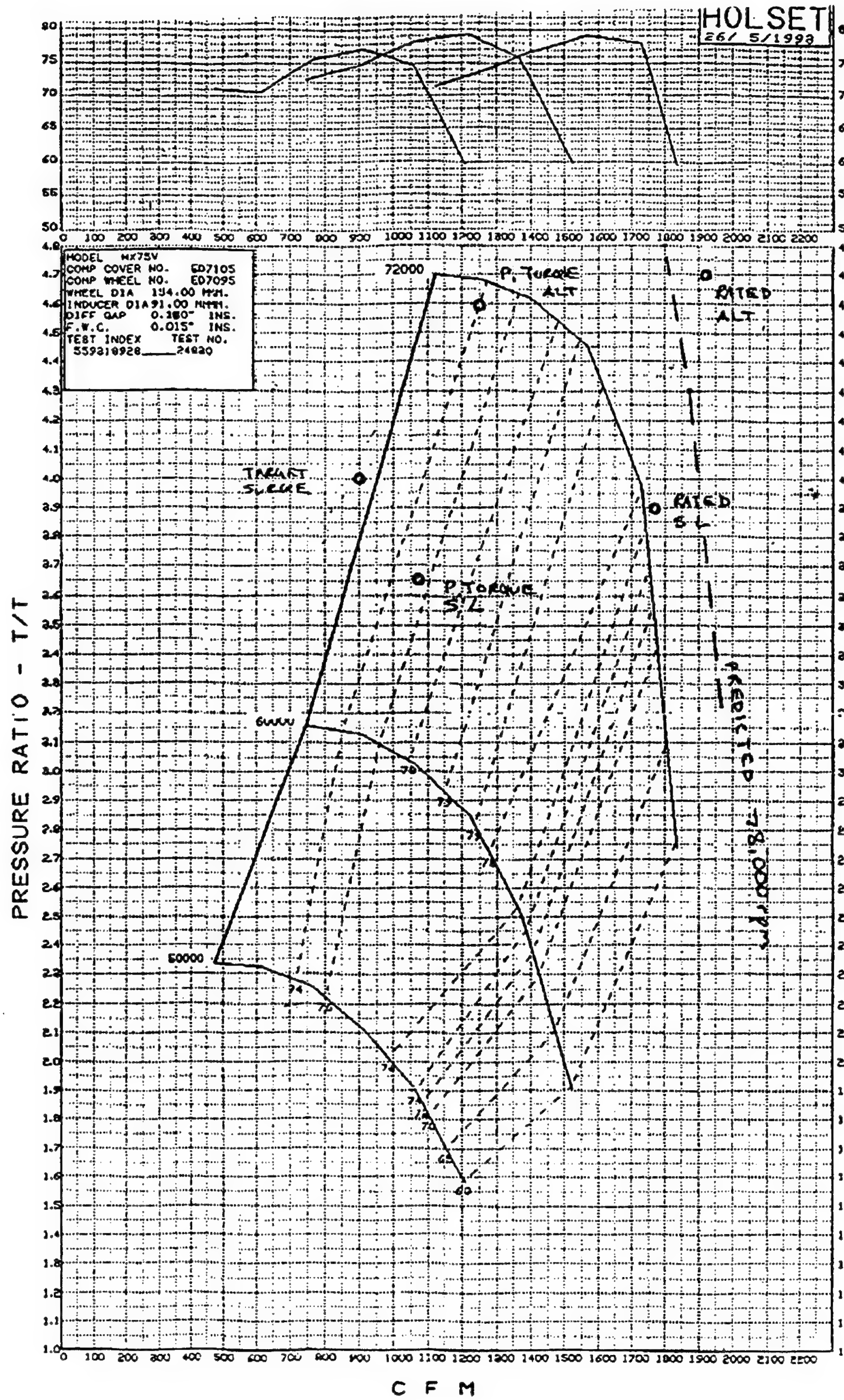


Figure VII-12. Compressor map

APPENDIX IX

Research Suggestions Considered

(Note: Appendix VIII is not used.)

APPENDIX IX

RESEARCH SUGGESTIONS CONSIDERED

IX.2.2 Engine

Many configurations, especially when accessories are included, occur during engine development. As a result, different dimensions and ratings may be quoted at different development times for the same engine model. Because of this fact, there may be some discrepancies in the tables in this section. Nevertheless, the data are useful when considering engine compactness.

There are two general approaches to increasing system compactness. The first is to improve system integration, while the second approach increases the power density of the basic engine bulk and its components after a thorough thermodynamic analysis and cycle configuration study.

IX.2.2.1 System integration

TABLE IX-1 shows a compactness comparison for a number of advanced power packs. The increased importance of compactness has prompted increased attention to system integration. For example, note the low percentage of unused volume in the dAIPS in TABLE IX-1. During the AIPS program, integration was considered highly important. Significant progress was made in system layout and integration, and consequently in compactness.

TABLE IX-1. General Powertrain System Compactness Comparison

Power System	MTU 883	CV-12 Perkins	Poyaud Hyperbar	RPI Rotary	dAIPS
Design	90°V12	V12	UDV8X	4068R	60°V12
Status	Lab	Service	Lab	Lab	Lab
Component	Dunk Volume, cu. ft				
Engine	34.4	34.0	35.0	34.0	34.0
Transmission	35.0	35.0	35.0	35.0	34.9
Cooling	33.0	41.0	40.0	30.0	16.4
Airhandling	8.4	8.4	8.4	11.8	8.4
Exhaust	2.0	2.0	2.6	2.0	2.0
Fuel & handling sys	43.9	45.0	46.6	47.0	39.3
Battery/noise	18.0	18.0	18.0	18.0	18.0
Unused volume	40.3	40.6	40.3	34.2	17.0
Total system – Envelope vol.	**215.0**	**224.0**	**225.9**	**212.0**	**170.0**

This table shows the benefit of the dAIPS package, which has the smallest envelope volume (170.0 cu. ft). Superior system integration resulted in a low unused volume of 17.0 cu. ft. The compactness of the oil cooling system and the use of low heat rejection technology clearly made a significant difference in this configuration. It should be noted that for the compactness study, the (l × w × h) engine envelope volume is used while elsewhere the dunk value is used.

This comparison shows that the dAIPS design compactness is achieved mainly by reduced cooling requirements. In addition, there seems to be no disadvantage in using a high airflow engine such as Hyperbar insofar as air handling system size is concerned. The comparison also indicates the value of low heat rejection development and the potential gain in developing compact heat exchangers for engine cooling. For most engines, the cooling subsystem occupies the same amount of space as the basic engine itself.

The design process of fitting components into a vehicle-dictated shape may require component size changes, and may therefore affect component performance. Consequently, iteration between the design management and an analytical model of the powerplant system must include the effects of these changes in order to truly optimize the system design. An iterative optimization process beginning from a clean layout would be required. For the dAIPS engine compartment layout, the unused volume is only 17 cu. ft. This is lower than the unused volume in the other engines. Some of this unused volume could be saved in a new engine compartment layout. It is clear that major volume envelopes are needed for the basic engine, cooling system, and fuel handling system.

For this report, it was only possible to concentrate on the basic engine layout and to study whether further compactness can be gained. Of course, it is also important to analyze whether the gain in basic engine compactness is not reduced by an increase in cooling equipment and/or air handling or fuel handling subsystems. This can only be ascertained by making a series of computer engine layouts. The final system integration for compactness can only be made when a "best we can do" basic engine configuration is projected. This projection will be presented in the next few sections.

a. <u>Basic engine compactness study</u>

This section deals with the potential for increased compactness by engine layout parameter optimization. When an engine cubic inch displacement (*CID*) is selected, there are several layout parameters that can be varied to achieve maximum compactness. These are as follows:

- number of cylinders, n
- bore size, B
- bore-stroke ratio, B/S
- Vee angle, θv

Engine compactness can be increased and engine specific weight can be reduced by using small bore engines and having many cylinders [Taylor's Internal Combustion Engine (ICE) text]. Simon Chen and Tang Wu projected in the 1960s that a 4-in. stroke engine could be 30 percent lighter than a 6-in. stroke engine at the same brake mean effective pressure (BMEP) on a lb/hp basis. This projection can be seen in Fig. IX-1.

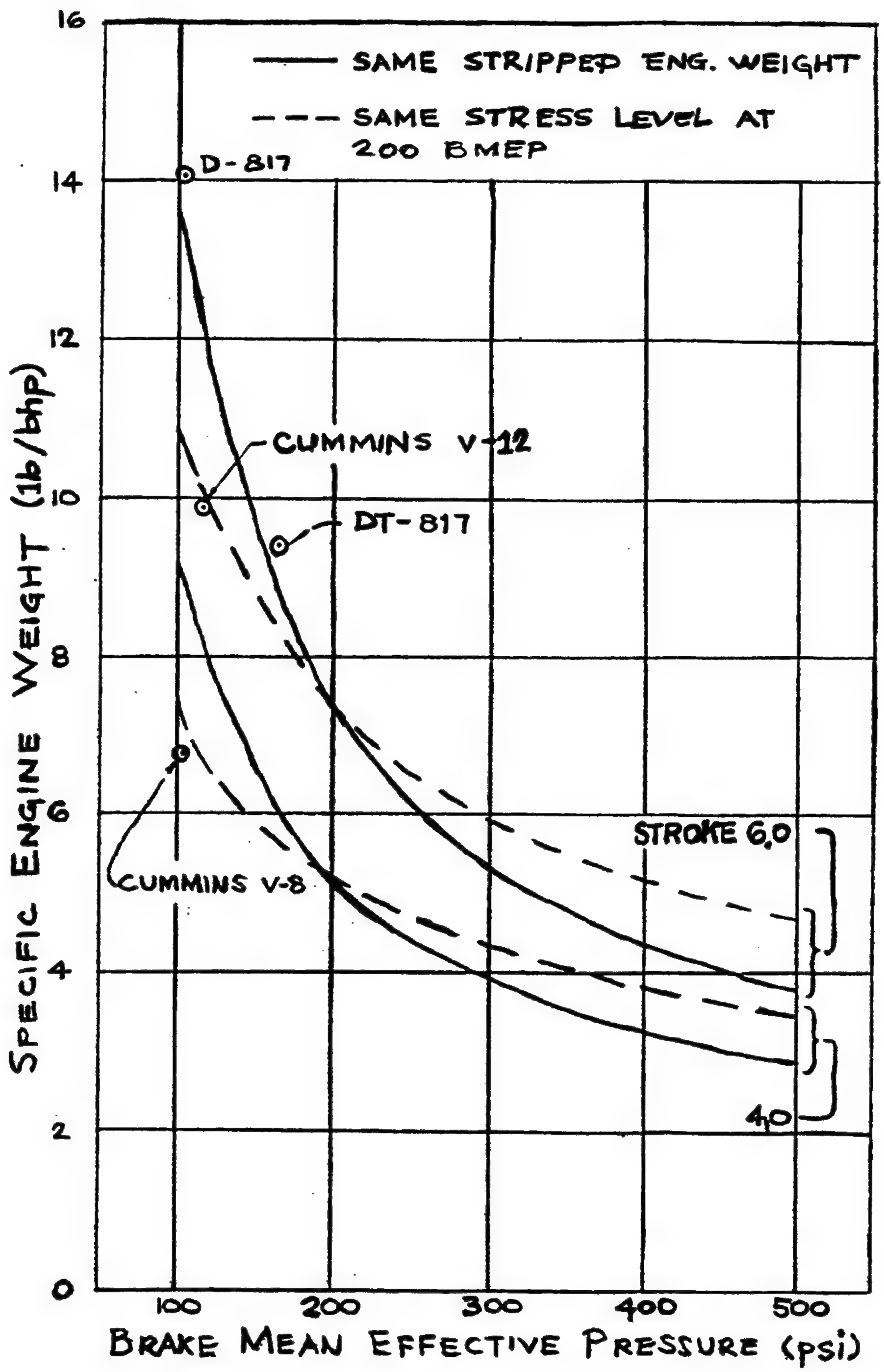


Figure IX-1. Projection for high BMEP engine weight (4- and 6-in. stroke engines)

The objective of the ensuing study is to find the optimum range for the number of cylinders, bore size, bore-stroke ratio, and Vee angle in order to attain the highest feasible ratio of engine displacement volume (i.e., *CID*) over the engine envelope volume (*EV*)--defined as (l × w × h) in cubic feet. The following design parameter ranges have been selected to limit the extent of the study:

- Design parameters

  n = 8 to 16 (number of cylinders)
  B/S = 1.2 to 0.8 (bore-stroke ratio)
  θv = 60°, 90°, 120° (Vee angle)

In addition to the above design parameters, the engine is limited by the following development parameters:

- Development parameters

  fpm = 2,500 to 2,800 ft/min (piston speed)
  Pmax = 2,400 to 3,000 psi (maximum cylinder pressure)
  Tgw = 450 to 520°F (mean gas side wall temperature index)

In general, basic engine design would need these parameters as inputs to design an engine with reasonable mechanical efficiency and fatigue life.

b. Engine displacement – envelope volume ratio

The purpose here is to lay out an engine configuration which will provide minimal engine envelope volume (*EV*), defined as follows:

- $$EV = H \cdot L \cdot W \qquad \text{(Eq. 1)}$$

Using the MTU 883 engine as a baseline, the maximum *CID/EV* parameters can be derived by using the number of cylinders (n), bore-stroke ratio (B/S), and Vee angle (θv) as independent variables. Engine height, width, and length are functions of these variables. The bore size is determined based on an assumed BMEP and piston fpm. The engine *CID* is a constant for this analysis.

B/S, n, and θv are the independent design variables. The equation could be used to find the minimum *EV* for the range of variables used. The MTU engine configuration is used as the baseline to determine the coefficient in Equations 2a, 2b, and 2c.

$$\bullet \quad \frac{CID}{EV} = \frac{constant}{H \cdot L \cdot W} \tag{Eq. 2}$$

$$H = (2.26 \cdot S + 2.1 \cdot B)\cos\theta v + 1.6 \cdot S \tag{Eq. 2a}$$

$$W = 2 \cdot (2.26\ S + 2.21 \cdot B)\sin\theta v + 0.53 \cdot B \tag{Eq. 2b}$$

$$L = (1.36 \cdot 0.5 \cdot nB + 3.0)B \tag{Eq. 2c}$$

A study of Equation 2 shows the following:

- A 16-cylinder version (with a 4.828-in. bore size) would reduce envelope volume by approximately 25 to 30 percent. The MTU 883 baseline is a V-12 configuration with a 5.669-in. bore size.

- A narrow Vee engine assists in compactness. The envelope volume of the MTU engine could be reduced by 16 percent if the 90° Vee angle were reduced to 60°. The dAIPS engine design has a narrow Vee angle, which results in a narrow engine width.

- For the MTU configuration, some minor envelope volume reduction could be achieved by using a larger bore-stroke ratio, around *B/S* = 1.1, but the reduction of envelope volume is only 3 percent.

This study confirms that the dAIPS engine layout has a near optimum *CID/EV* ratio. However, further gain can be made through the design of an engine family with approximately 100 hp/cylinder and with a bore size of 4.828 in. rather than 5.669 in. A more compact V-16 version with a bore of 4.828 in. could replace the dAIPS V-12 at the same level of horsepower.

c. Maximum hp density study

For a fixed displacement, the engine hp is a linear function of BMEP, piston speed, and bore area. Horsepower density (hp/*CID*) is a function of BMEP and rpm and is inversely proportional to the stroke length.

$$\bullet \quad HP = CID \cdot BMEP \cdot RPM \cdot constant$$

$$= CID \cdot BMEP \frac{FPM}{S} \cdot constant$$

$$\frac{HP}{CID} = BMEP \cdot \frac{FPM}{S} \cdot constant \tag{Eq. 3}$$

$$HP = B^2 \cdot BMEP \cdot FPM \cdot constant \tag{Eq. 3a}$$

d. Piston speed increase potential

Engine speed is limited by excessive inertia load and engine friction. MTU 883 is the leader in high speed development, as it runs at a piston speed 25 percent higher than that of the dAIPS engine (see TABLE IX-2). The MTU engine's fuel performance is not adversely affected by high speed operation. Its engine is quite advanced in its mechanical design and development.

TABLE IX-2. Bore-Stroke Ratio

	Bore	Stroke	*B/S*	rpm	fpm*
MTU 883	5.669	5.512	1.02	3,000	2,756
Perkins V-12	5.315	5.984	0.90	2,400	2,394
dAIPS	5.906	5.118	1.16	2,600	2,218
Poyaud Hyperbar	5.591	5.118	1.11	2,500	2,133

* ft/min

Operating at an rpm that is 25 percent higher than the present rated speed would require lengthy proof testing and design refinements. It would be simpler to increase the dAIPS engine stroke by 25 percent while keeping the rpm constant. A bore-stroke ratio in the neighborhood of 0.80 to 0.90 is quite common for commercial engines. This would require a new layout and an increase in engine height, a consideration for future high-performance engines.

e. Engine benchmarking

The engine performance parameters of the dAIPS engine, MTU 883, Perkins CV12, and Poyaud Hyperbar diesels are compared in TABLE IX-3. Engine envelope volumes are calculated from length, width, and height data shown.

The Poyaud Hyperbar performance data are from RD & E Technical Report No. 13537, authored by A. Lemmo (see Ref. 4). All other data are from "Diesel Engine Characteristics," TABLE IX-4. Other useful tank engine characteristics are shown in TABLES IX-5 and IX-6.

dAIPS Engine – In terms of compactness, the dAIPS is superior (*CID/EV* = 36.9). It has the smallest EV and the narrowest width due to its small Vee angle (60°) and superior packaging. This superior packaging and performance was confirmed earlier in Section IX.2.2.1.b. Further reduction of envelope volume on the order of 25 to 30 percent could be feasible if a V-16 layout is used together with a smaller bore.

TABLE IX-3. Engine Benchmarking

	dAIPS	MTU 883	Perkins CV12	Poyaud Hyperbar
hp/rpm	1,450/2,600	1,474/3,000	1,500/2,400	1,500/2,500
Length, in.	68.9	66.0	56.6	54.1
Width, in.	30.6	37.4	55.8	57.5
Height, in.	37.4	33.3	45.2	35.6
EV, cu. ft	45.6	47.6	82.6	64.1
CID, cu. in.	1,682	1,670	1,593	1,005
EW, lb	4,170	3,638	4,500	4,750
hp/*EV*	31.8	31.0	18.1	23.4
lb/hp	2.88	2.47	3.00	3.16
CID/EV	36.9	35.0	19.3	15.7
BSFC, rated power	0.32	0.362	0.393	0.381
BMEP, rated power	263	233	311	473
BMEP, peak torque	315	272	348	532
Mean piston speed, fpm	2,218	2,756	2,394	2,133

MTU 883 – MTU has the lowest engine height due to its 90° Vee configuration. It also has the lowest specific weight, at 2.47 lb/hp, showing design sophistication and packaging superiority despite its rather long 5.512 stroke (which is 8 percent longer than that of dAIPS). Design and development sophistication is also demonstrated by its high speed ratings (2,756 fpm and 3,000 rpm).

Perkins CV12 – The CV12 has a 60° Vee angle similar to the dAIPS engine. Despite this narrow Vee angle, the engine is not as well packaged as the dAIPS, resulting in an engine envelope volume that is 80 percent larger (82.6 cu. ft versus 45.6 cu. ft). The major difference is in engine height, which is affected by a long engine stroke of 5.984 (17 percent bigger than the dAIPS engine). The engine length is fine due to the smaller bore size. The developed BMEP is 311 psi at rated, and 348 psi at peak torque. This moderately high BMEP was developed with a modest peak cylinder pressure (Pmax = 2,200). The low cylinder pressure performance (152 bar) shows Perkins' strength in turbocharging (PR = 4), aftercooling (A-A) and combustion (*F/A* = 0.04). The weight of this engine is slightly high.

Poyaud Hyperbar – For this engine, energy input to the exhaust turbine is greatly increased through the use of an auxiliary burner. This results in a high inlet pressure ratio in the order of 6:1 (compared to 3.1 to 3.5 for the others) and a max air consumption rate of 7 lb/s (as compared

TABLE IX-4. Diesel Engine Characteristics

Engine Parameters	Engine: dAIPS	MTU 883	Perkins CV12	Poyaud Hyperbar
Number of cylinders	12	12	12	8
Cylinder configuration	60°V	90°V	60°V	90°V
Bore, in.	5.906	5.669	5.315	5.591
Stroke, in.	5.118	5.512	5.984	5.118
Displacement, cu. in.	1,682	1,670	1,593	1,005
Compression ratio	15.0	14.0	10.5	7.8
Max. power (hp) at rpm	1,450/2,600	1,500/3,000	1,500/2,400	1,500/2,500
Max. torque (lb-ft) at rpm	3,498/1,560	2,860/2,100	3,680/1,700	3,370/2,000
Mean piston speed, ft/min	2,218	2,756	2,394	2,133
BSFC at rated power, lb/hp-hr	0.320*	0.369	0.393	0.381
BMEP at max. power, psi	263	237	311	473
BMEP at max. torque, psi	315	272	348	532
Power/displacement, hp/cu. in.	0.86	0.88	0.94	1.49
Power/piston area, hp/in.^{2}	4.41	4.95	5.63	7.64
Peak cylinder pressure, psi	2,600	2,000	2,200	2,300
Injection pressure, ksi	23	16	12	14
Max. fuel consumption, lb/hr	464	534	600	570
Air-fuel ratio	28	27	25	40
Max. air consumption, lb/s	3.86	3.75	4.2	6
Turbocharger pressure ratio	3.5	3.1	4	6
Intercooling	A-O	A-L	A-A	A-L
Engine coolant	Oil	Water	Water	Water
Max. coolant temperature, °F	340	225	239	220
Heat rejection, Btu/hp-min	19	31.5	27	28
Cooling fan power, hp	120	240	173	190
Length, in.	68.9	66.0	56.6	54.13
Width, in.	30.6	37.4	55.8	57.56
Height, in.	37.4	33.3	45.2	35.63
Weight, lb	4,170	3,638	4,500	4,750

* Target

TABLE IX-5. Tank Powerpack Weight Estimates, lb

Component	Diesel AIPS	MB 873	CV-12	MTU 883	Mack Hyper	Poyd Hyper	AVCR 1360	AGT 1500	TMEPS AGTT 1500	Turbine AIPS	Rotary
	Engine Model										
Engine	4,170	5,711	4,500	3,638	3,800	4,750	4,550	2,500	2,600	2,100	2,200
Transmission (X1100)*	4,000	4,350	4,350	4,350	4,350	4,350	4,350	4,350	4,600	3,900	3,800
Fuel (one BFD)	1,763	2,300	2,100	2,178	2,400	2,400	2,300	3,500	2,800	2,045	2,600
F tanks (one BFD)	200	260	240	250	270	270	260	400	320	300	290
Electr (alt and harn)	140	150	150	150	150	150	150	170	200	50	150
Ancillaries (hyd, contr, etc.)	220	300	300	300	300	300	300	300	300	445	300
Exhaust	50†	50	50	50	50	50	50	170	170	442†	50
Final drives	1,880	1,934	1,934	1,934	1,934	1,934	1,934	1,934	1,934	1,880	1,934
Cooling system (fans, dr. rad. duct)	580	1,463	1,297	1,297	1,297	1,297	426	596	750	205	1,000
Air cleaner	180	200	200	180	200	200	200	350	350	450	200
Batteries @ 80 lb ea.	480	640	480	480	480	480	640	480	320	480	480
Grilles	226	1,295	989	989	1,295	1,295	989	0	0	–230	660
Coolant	116	754	524	460	449	450	0	0	0	0	300
Oil-engine	200	210	180	180	170	170	240	50	50	30	100
Oil-transmission	360	300	300	300	300	300	300	300	300	369	300
APU	--	--	--	--	--	--	--	--	500	--	--
	---------	---------	---------	---------	---------	---------	---------	---------	---------	---------	---------
Total weight	14,565	19,917	17,594	16,736	17,445	18,396	16,689	15,100	15,194	12,466	14,364
Total increase over M1A1	–535	4,817	2,494	1,636	2,345	3,296	1,589	Ref	94	–2,634	–736

* X1100 at 4,350 lb
† Without IR suppression

TABLE IX-6. Tank Engine Characteristics

	Engine Model												
Parameter	Diesel AIPS	MB 873	MB 871	CV-12	CV-12	MTU 883	Mack Hyper	Poyd Hyper	AVCR 1360	AGT 1500	AVDS 1790	AVDS 1790	Turbine AIPS
Horsepower	1,450	1,474	1,179	1,200	1,500	1,500	1,500	1,500	1,500	1,500	900	1,200	1,408
No. of cylinders	12	12	8	12	12	12	8	8	12		12	12	
Type of cooling	Oil	Liquid	Liquid	Liquid	Liquid	Liquid	Liquid	Liquid	Air		Air	Air	
Displacement	1,682	2,909	1,939	1,593	1,593	1,670	998	1,005	1,360		1,790	1,790	
Bore	5.91	6.693	6.693	5.315	5.315	5.67	5.375	5.591	5.375		5.75	5.75	
Stroke	5.12	6.89	6.89	5.984	5.984	5.51	5.5	5.118	5.0		5.75	5.75	
Compression ratio	15	18.0	16.0	12.0	10.5	14.0	7.8	7.0	8.16		14.5	13.0	
Rated rpm	2,600	2,600	2,600	2,300	2,400	3,000	2,500	2,500	2,600	3,000	2,400	2,400	24,400
BMEP at rated rpm	263	157	189	259	311	237	476	473	336		166	221	
hp/cu. in. disp. vol.	0.86	0.516	0.619	0.75	0.94	0.90	1.5	1.49	1.10		0.502	0.67	
hp/in^2 piston area	4.41	3.55	4.26	4.51	5.63	4.95	8.26	7.64	5.51		2.89	3.85	
Piston speed (rated), fpm	2,218	2,990	2,990	2,294	2,394	2,756	2,292	2,133	2,167		2,300	2,300	
BSFC at rated	0.320*	0.414	0.414	0.364	0.393	0.369	0.380*	0.381	0.420	0.50	0.39*	0.39	0.36
Length, in.	68.9	66	50	56.6	56.6	66	54	50	73	67	68	68	55
Width, in.	30.6	76.8	76.8	55.8	55.8	37.4	53	44	60	39	75	75	34
Height, in.	37.4	43	43	46.5	45.2	33.3	43	39	46	32	45.5	45.2	40
Weight, lb	4,170	5,711	4,750*	4,358	4,500	3,638	3,800	4,750	4,550	2,500	4,900	4,900*	2,100

* Estimated

to 3.86 for dAIPS). The use of an auxiliary burner to reach high BMEP conditions seems effective, although the controls and engine design are rather obsolete. Despite this high BMEP, the engine package and weight are not very attractive due to the following:

- 8-cylinder layout – A Vee 12 layout would reduce the engine length by 15 to 20 percent, and the engine envelope volume by 30 to 45 percent.

- 90° Vee angle – A 60° Vee angle would reduce the engine width or the engine envelope volume by 16 percent.

- The engine packaging is not sophisticated.

- A very conservative piston speed of 2,133 ft/min is used.

The Poyaud Hyperbar design has poor fuel economy due to two basic factors:

- Low compression ratio – The use of a 7.8:1 compression ratio is effective in limiting the peak cylinder pressure to 2,300 psi. This is an amazingly low figure for 532 BMEP; however, a low expansion ratio of 7.8:1 is bad for fuel economy.

- Auxiliary burner – The burner is effective for creating high pressure turbocharging; however, it also creates high idling fuel consumption. It is necessary to turn on the burner to start the engine for this particular design. This is a drawback due to the low compression ratio design.

This benchmarking shows that a further gain in power density for any of the present configurations will be limited to 20 to 30 percent, based on current world-class technology. The benchmarking also indicates that 2,756 ft/min and a rated BMEP level of 473 psi are feasible. The question remains whether a future engine could achieve both of these development objectives. If both objectives can be realized, a future engine based on the dAIPS engine basic design could have a power density of 40 to 45 hp/cu. ft, provided the thermal load limit is under control. If only one of the objectives, either high BMEP or high piston speed, can be realized, a power density in the order of 40 to 58 hp/cu. ft can be reached. This is based on current "best of class" technology. The dAIPS engine could be rated at 2,000 hp.

Basically, BMEP is directly proportional to density ratio (ρ/ρ_o or boost level), fuel-to-air ratio (*F/A*), and engine efficiencies ($\sum Ee$ in Eq. 5). The BMEP/$\rho/\rho_{\rho o}$ term is a function of *F/A* and BSFC (Eq. 5a). This term can also be expressed in brake specific air consumption (BSAC in Eq. 5b). BMEP/ρ is actually proportional to hp-hr/air rate, or the effectiveness of displacement utilization and air combustion. TABLES IX-4 through IX-6 show various diesel and tank engine characteristics. These data are helpful in benchmarking.

IX.2.2.2 Power density and BMEP potential

Engine compactness (hp/*EV*), or power density, is directly proportional to BMEP. High BMEP is the most fruitful development route available, as further increases in piston speed beyond

2,756 fpm will be quite limited. The power density hp/*EV* and BMEP potentials are expressed as follows:

- $$\frac{HP}{EV} = \frac{CID}{EV} \cdot BMEP \cdot \frac{FPM}{S} \cdot constant \qquad \text{(Eq. 4)}$$

- $$BMEP = \frac{\rho}{\rho_o} \cdot \frac{F}{A} \cdot \sum Ee \cdot constant \qquad \text{(Eq. 5)}$$

$$\frac{BMEP}{\frac{\rho}{\rho_0}} = \frac{F}{A} \cdot \frac{1}{BSFC} \cdot constant \qquad \text{(Eq. 5a)}$$

or

- $$\frac{BMEP}{\rho} = \frac{1}{BSAC} \cdot constant \qquad \text{(Eq. 5b)}$$

ρ/ρ_o = density ratio
F/A = fuel/air ratio
$\sum Ee$ = product of engines' efficiencies
= $Eind \cdot Emech \cdot Epump$
$BSAC$ = Brake specific air consumption
= air (lb)/hp/hr

Barriers to BMEP increases – The amount of BMEP increase that will be feasible depends on whether certain mechanical, thermal load, turbocharging, and injection problems can be resolved through design upgrades and systematic development. If the efficiencies are not deteriorating, BMEP per unit density or BMEP over ρ/ρ_o should increase linearly with *F/A* until it hits the stoichiometric ratio. BMEP per unit density is simply a function of the fuel-air ratio and engine cycle, combustion, and mechanical efficiencies. The design and development projects needed to reach a BMEP of 350 to 400 include the following:

- turbocharger performance development for a high pressure ratio (4:1 - 5:1) compressor capable of operating at wide flow range and high turbo efficiency. This is required to produce the density ratio (at inlet) necessary for high BMEP engines with reduced aftercooling.

- mechanical design limit extension to cope with high cylinder pressure. The engine components needing improvement will include an articulated piston, improved bearing design, robust crankshaft, viscous damper, etc.

- thermal limit extension to cope with the high cylinder temperature associated with high BMEP and high inlet temperature conditions. The engine components affected by high thermal load include the cylinder head fire deck, exhaust valve and valve seat, injector nozzle, piston crown and combustion bowl, top compression ring, ring-sleeve interface, etc. Thermal design limits can be extended in two ways: through improved material or by enhanced local cooling.

Successful High BMEP Configuration – A successful high BMEP configuration will have the following characteristics:

- an advanced design turbocharger to provide a high degree of exhaust energy recovery within a wide operating range;

- advanced power components with advanced local cooling for reliable operation at the high cylinder pressure and high top ring groove temperature (Ttrg) conditions; and

- a flexible high pressure, advanced electronic injection system to provide good combustion performance at all operating conditions, including retarded injection timing.

Research and Development Required – Engine design and components suitable for high BMEP operation can be developed in successive steps as follows:

- Advanced high BMEP cycle analysis to select the best suited compression ratio, turbocharging ratio, degree of intercooling/aftercooling, degree of heat insulation, etc. The computer analysis should also provide thermodynamic, mechanical, and thermal stress predictions.

- Aggressive research on lightweight, high-temperature, low thermal stress engine materials.

- Aggressive research on a single cylinder test engine for advanced component development aimed at finding solutions to high temperature tribological (i.e., Ttrg, etc.) bottlenecks.

- Advanced research and development on the Hyperbar controls and on turbocharging controls for high pressure ratio applications.

- Advanced development in air handling, filtering, and aftercooling equipment for compactness and performance.

- Advanced development on low heat rejection (insulated) components for durability and reliability.

a. High BMEP cycle analysis

An increase in power density (or compactness) by 50 percent (from the current 31.8 hp/*EV* level) will require an increase in BMEP by at least 50 percent, assuming piston speed remains the same. This increase could be achieved through a combination of the following:

- inlet density ratio (ρ/ρ_o) increase (boost increase);
- fuel-air ratio (*F/A*) increase (rich burning).

The objective of using the high BMEP cycle analysis is to determine the most suitable engine cycle parameters (variables of choice) to achieve the BMEP, fuel efficiency, and durability goals. It is much more affordable to study a computer engine than it is to develop a new test engine by trial and error.

- $$BMEP = \frac{\rho}{\rho_o} \cdot \frac{F}{A} \cdot LHV \cdot \sum Ee \qquad \text{(Eq. 6)}$$

- $$\sum Ee = Eisen \cdot Eadia \cdot Ecomb \cdot Epump \cdot Emech \qquad \text{(Eq. 7)}$$

- $$Eisen = 1 - \left(\frac{1}{CR}\right)^{K-1} \qquad \text{(Eq. 8)}$$

The compression ratio, expansion ratio, turbocharger boost ratio, degree of cooling or heat insulation, *F/A* ratio and combustion rate are "independent cycle variables" for the computer engine. Cylinder pressure, average gas side wall temperature, engine efficiencies, and BMEP are cycle results.

At this writing, a TACOM-specific, high BMEP cycle study has not been started. It is only practical, at the moment, to perform a preliminary parametric study using a set of existing parametric figures, Figs. IX-2 through IX-7. Because the methodology and the software were developed thirty years ago, the results of this study will be used in relative rather than absolute terms. At that time, we were concerned to reach 200 to 300 BMEP; today, our goal is raised to 400 BMEP and up.

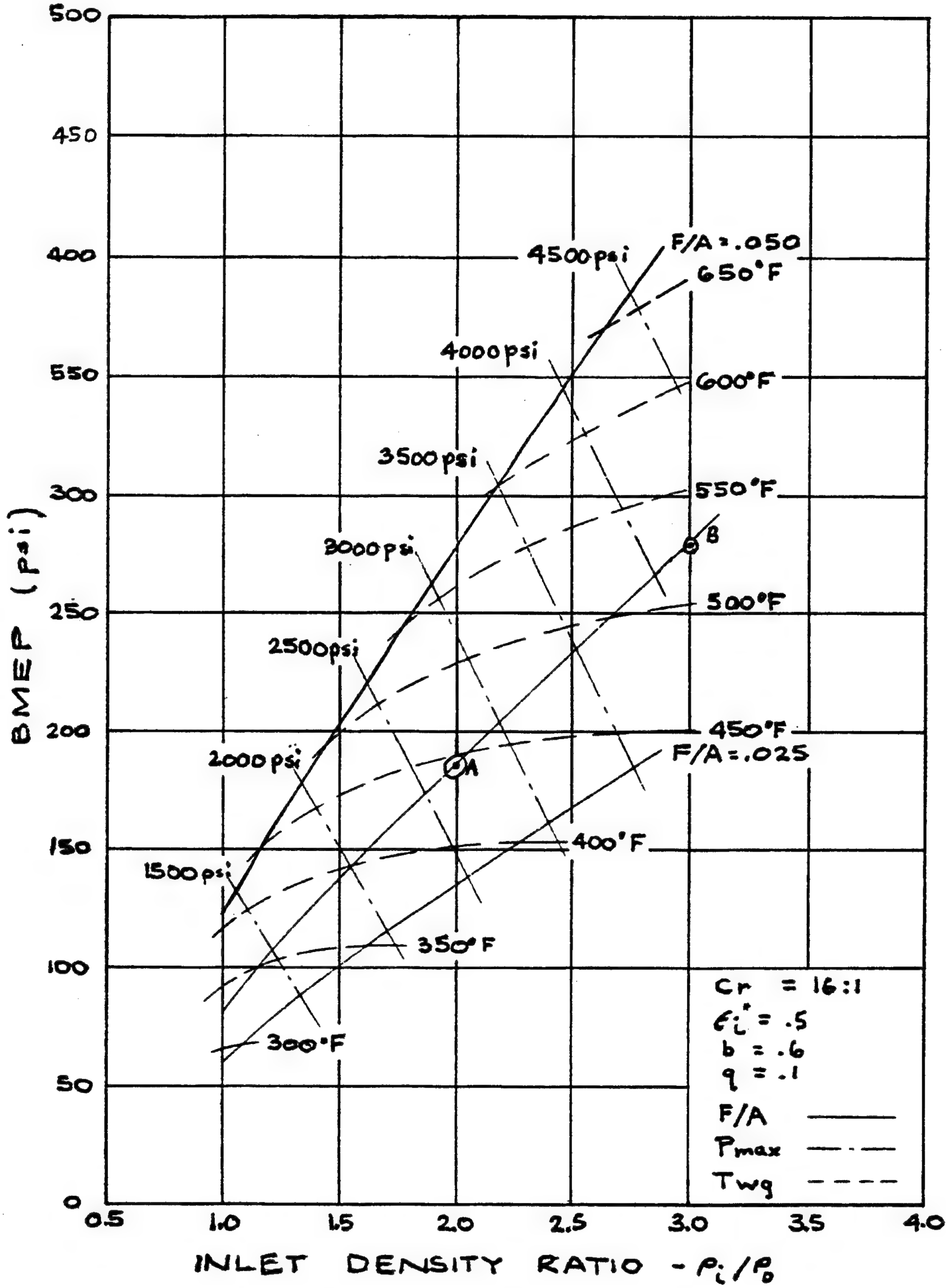


Figure IX-2. CR = 16, moderate aftercooling

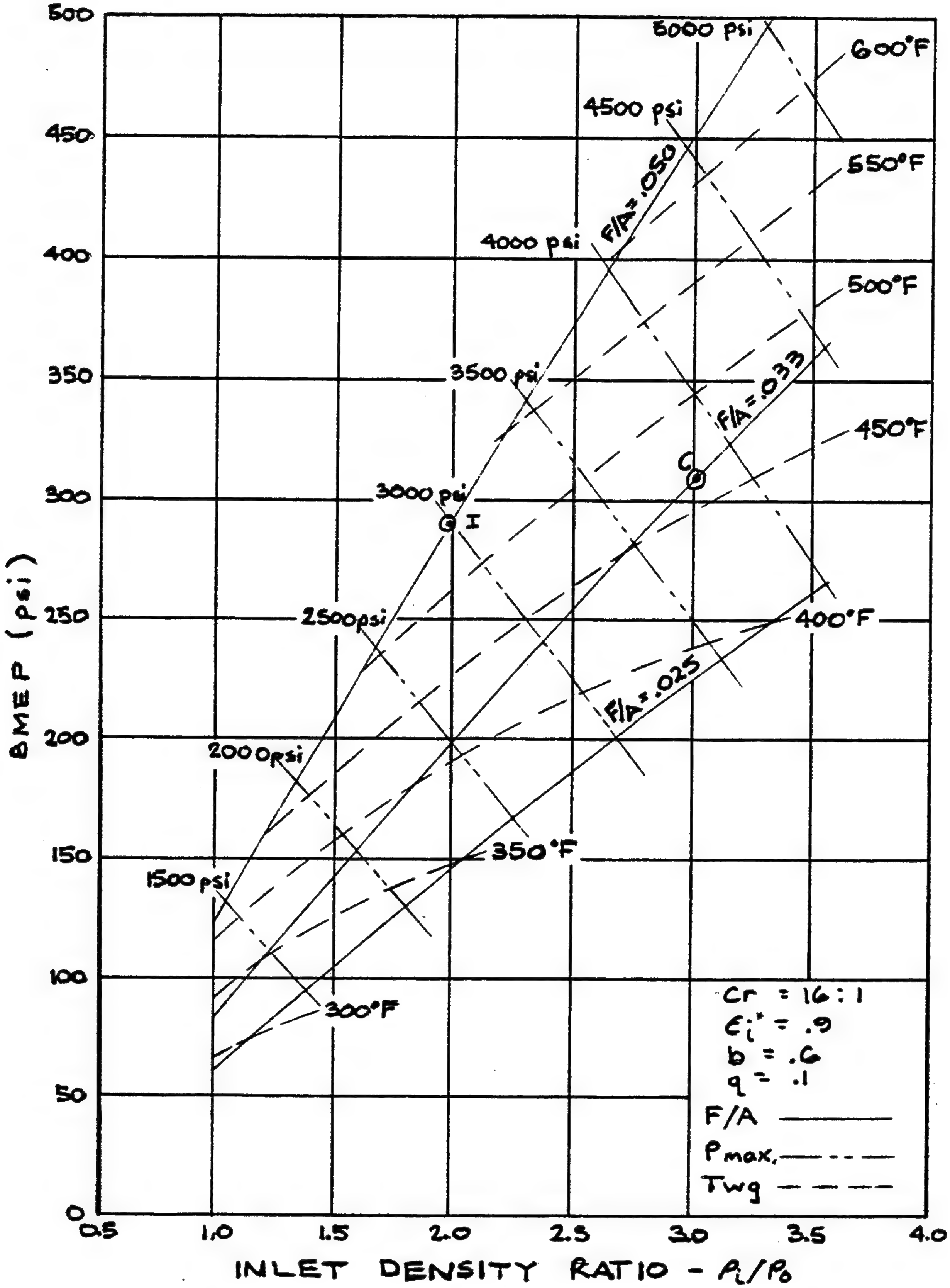


Figure IX-3. <u>CR = 16, high aftercooling</u>

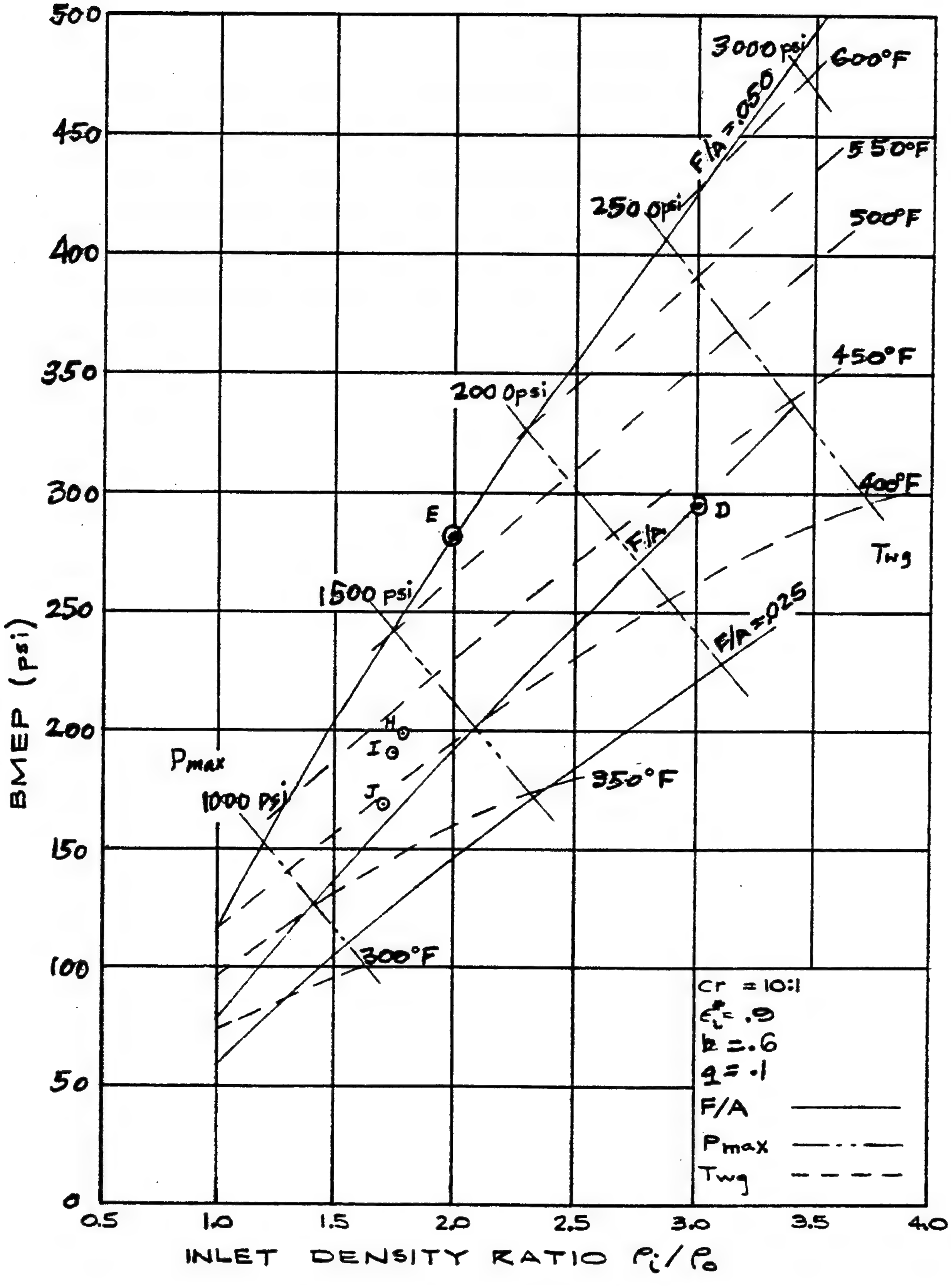


Figure IX-4. CR = 10, high aftercooling

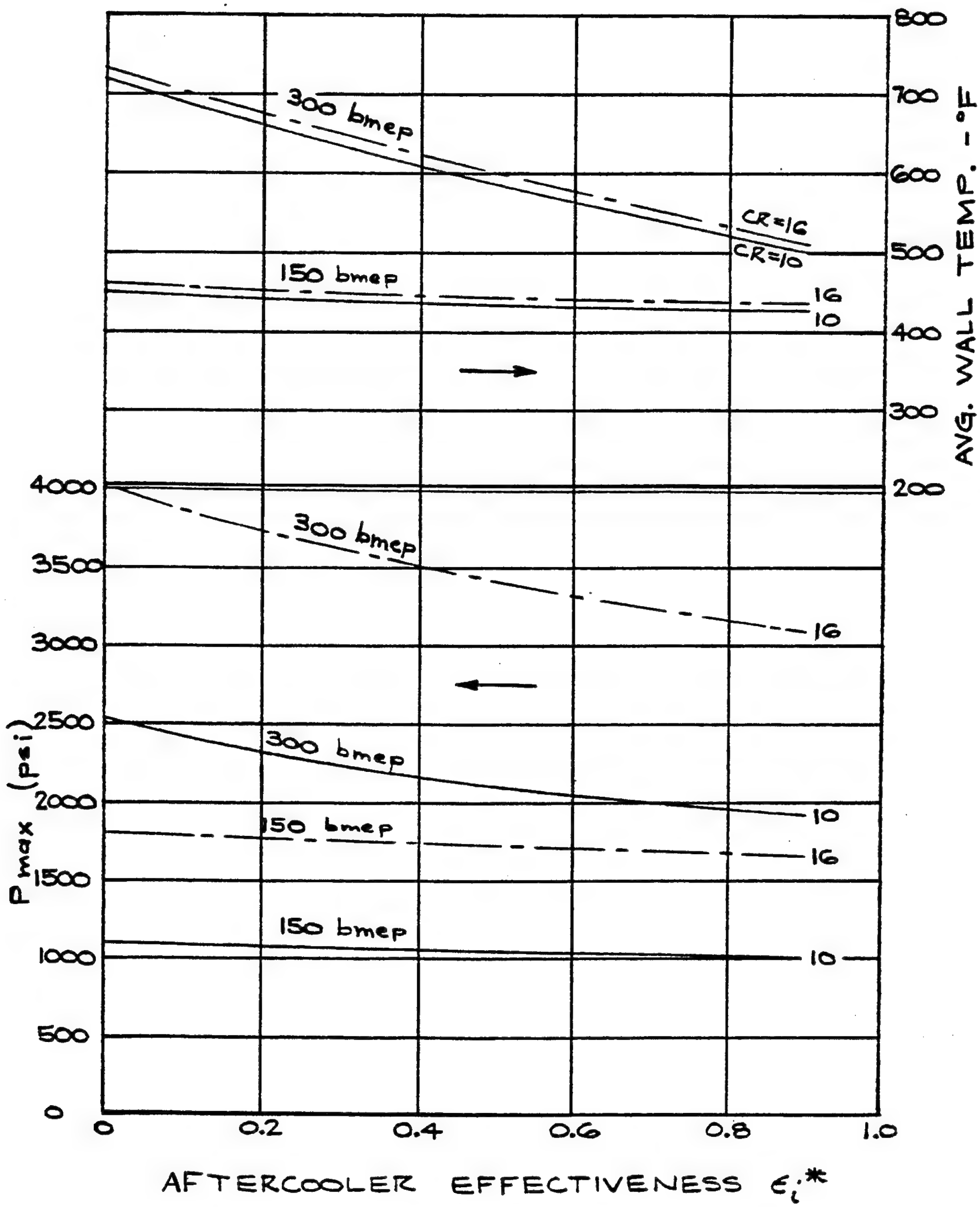


Figure IX-5. <u>**Effect of aftercooler effectiveness**</u>

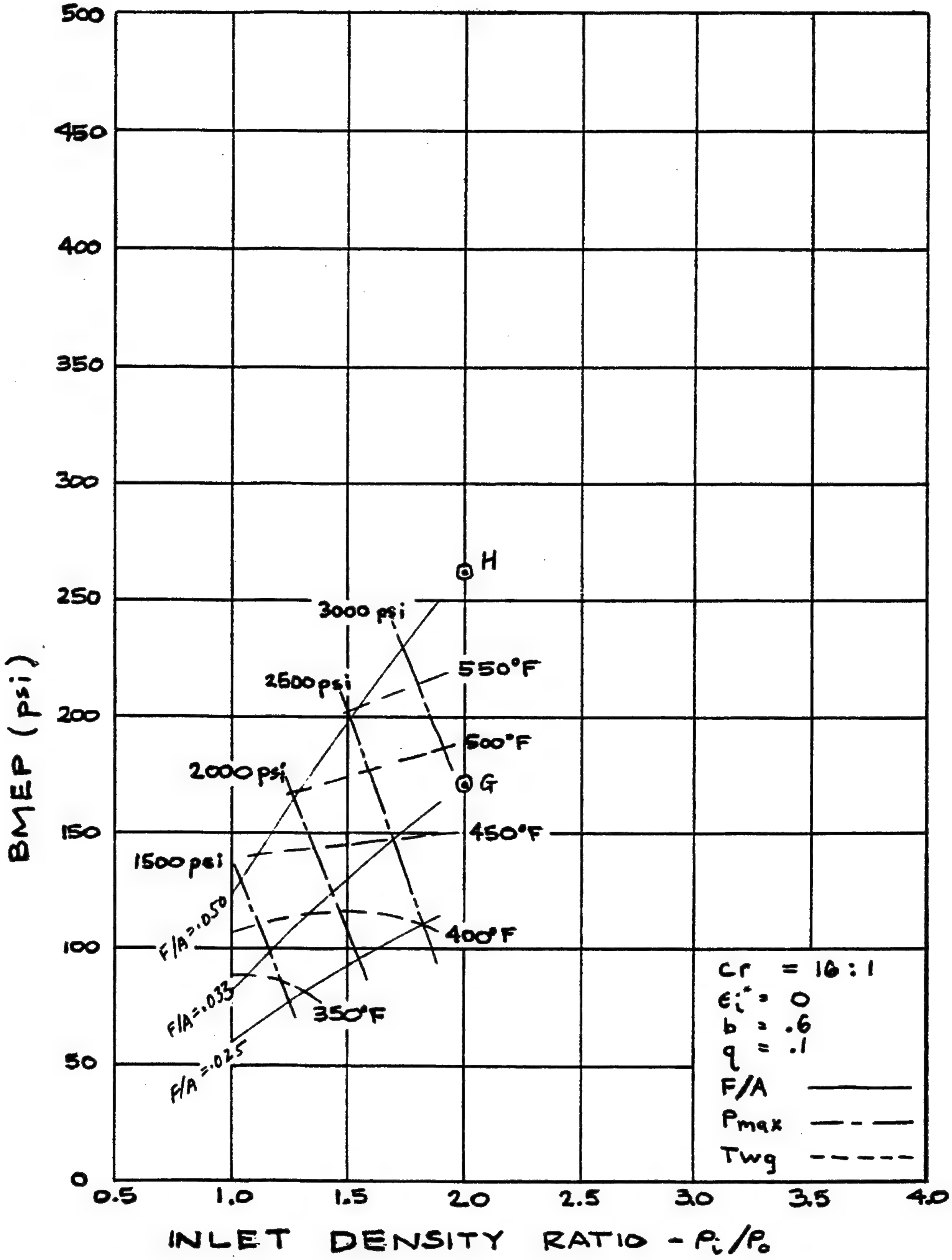


Figure IX-6. CR = 16, no aftercooling

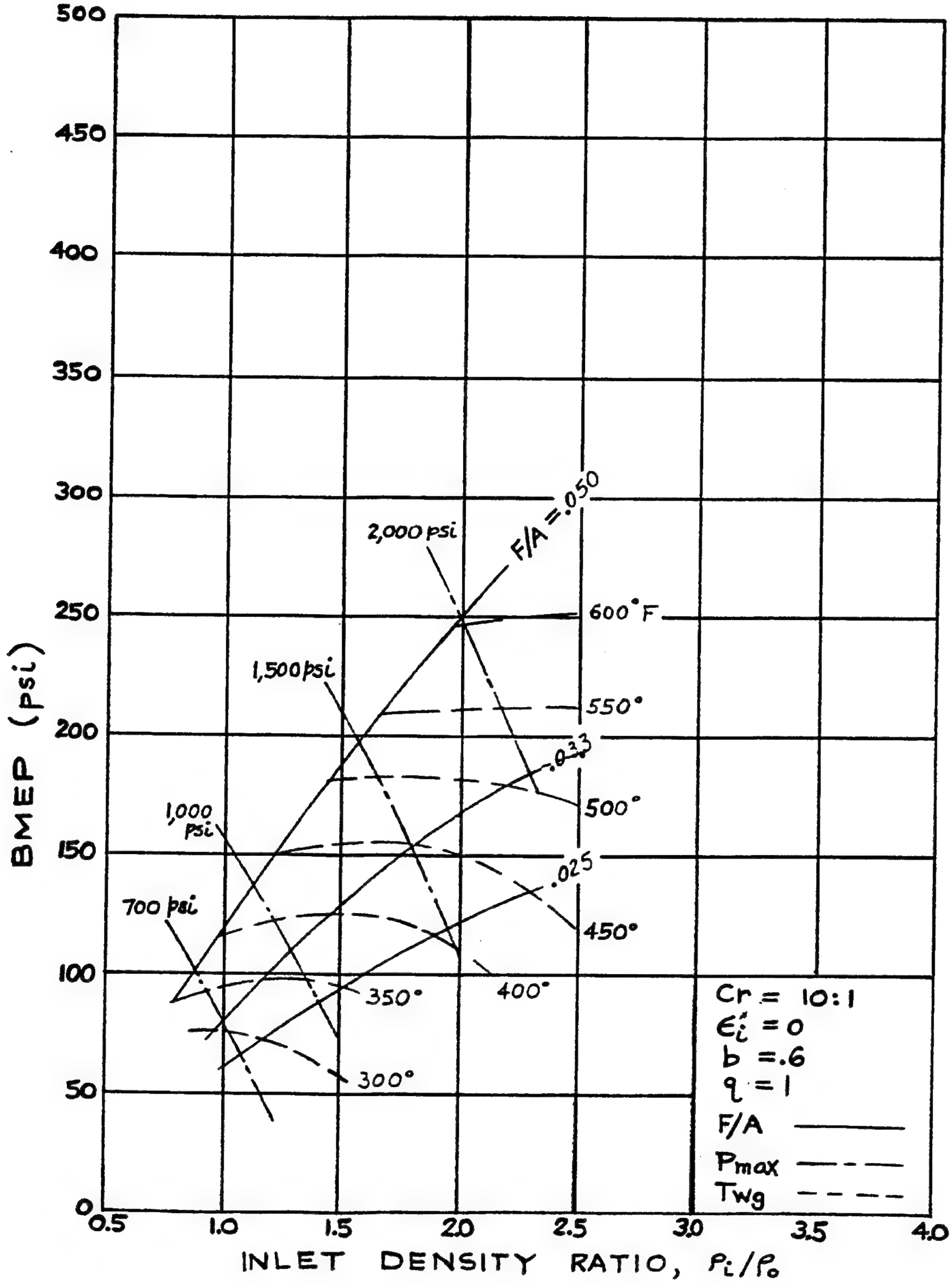


Figure IX-7. CR = 10, no aftercooling

b. Computer engine

The baseline computer engine (shown as Option A in TABLE IX-7) has the following characteristics:

- ρ/ρ_o = 2.0, BMEP = 190 psi, CR = 16, aftercooler effectiveness ε_i = 0.5 , *F/A* = 0.033 (Air-fuel ratio = 30:1);
- Pmax = 2,700 psi (rather high due to 60 percent constant volume combustion rate assumed);
- Twg = 440°F (based on modest aftercooling, ε_i = 0.5).

TABLE IX-7. Cycle Options

Case	ρ/ρ_o	*F/A*	ε_i	Pmax, psi	Twg, °F	CR	BMEP, psi	Figure
A	2.0	0.033	0.50	2,700	445	16	190	IX-2
B	3.0	0.033	0.50	4,150	525	16	280	IX-2
C	3.0	0.033	0.90	3,800	465	16	320	IX-3
D	3.0	0.033	0.90	2,200	435	10	295	IX-3
E	2.0	0.050	0.90	1,750	520	10	288	IX-3

300 ± BMEP could be achieved by several cycle options (see TABLE IX-7), including

- Aggressive turbocharging (Option B) increases the density ratio by 50 percent;
- Burning rich (Option E) increases *F/A* ratio by 50 percent;
- Increasing the density ratio and aftercooling effectiveness (Options C, D, and E) ε_i = 0.90, increased from 0.50; density ratio ρ/ρ_o is increased from 2.0 to 3.0;
- Reducing the compression ratio (Options D and E) from the 16:1 level to the 10:1 level for calculation of cylinder pressure (Pmax) and average gas side wall temperature (Twg).

Cases A, B, and C are computed with the baseline compression ratio (CR), which is 16:1. Compression ratio is equivalent to expansion ratio in this analysis. Cases D and E are computed with the CR reduced to 10:1. This comparison indicates that the prescribed limits Pmax < 3,000 psi, Twg < 520°F will be exceeded unless additional aftercooling is incorporated. Aftercooling is most effective in reducing average wall temperature (see Ref. 3, Fig. 9, p. 28). This correlation is reproduced in Fig. IX-5.

Case E indicates a potential "cycle route" to achieve high BMEP through rich burning. The fuel-air ratio is increased from 0.033 to 0.50 (a 50 percent increase). The barely tolerable Twg increase (ΔTwg = 75°F) is achieved by aggressive aftercooling ($\varepsilon_i = 0.90$). The inlet density increase ($\rho/\rho_o = 3.0$) is achieved through efficient turbocharging and aftercooling.

A combination of density ratio, fuel-air ratio, and aftercooling increases could produce the desired 50 percent increase in BMEP. A low compression ratio (CR = 10:1) and aggressive aftercooling could limit the mechanical load index (Pmax) and the thermal load index (Twg) to a tolerable level.

c. <u>The pros and cons of burning rich mixtures</u>

- $$BMEP = \frac{\rho}{\rho_o} \cdot \frac{F}{A} \cdot \sum Ee \qquad \text{(Eq. 9)}$$

The BMEP should increase linearly with *F/A* if engine efficiencies are not reduced by rich burning and if high mechanical and thermal loads are not out of control (see Fig. IX-8). If gasoline were allowed in a battle tank, the engine compactness could easily be enhanced by a factor of two. TABLE IX-8 shows weight and compactness data for a number of commercial and military engines. TABLE IX-8 shows that a Dodge V-10 has a power density (hp/cu. ft) of 27.72, while Arctic Cat, a small bore engine, has a power density of 69.04. If gasoline were used, the *F/A* ratio of 0.06:0.07 (16:1 *A/F*) range would be normal, and running at stoichiometric mixture would not cause heavy smoke and low combustion efficiency. But on-board gasoline fuel is not safe and is not recommended for this and other reasons.

The crucial question is how rich a diesel fuel/air mixture an engine can run efficiently without excessive smoke and deteriorating combustion. Figure 16 in Ref. 3 shows experimentally that the 60 percent smoke limit occurs at progressively lower *F/A* ratios as the density ratio is increased. This is reproduced in Fig. IX-9. This is 30-year old data, but it shows the combustion limit in terms of the burning rich limit of a conventional diesel fuel. Recent R&D indicates that an oxygenated fuel could reduce the smoke and emissions levels significantly. This would be a fruitful project for high power density development.

Case A in TABLE IX-9 is the baseline computer engine. Case H indicates a fictitious case of increasing fuel-air ratio from 0.033 to 0.50 with no aftercooling and at the same density ratio (no additional boosting) as the baseline engine. The Pmax is increased by 1,400 psi, and Twg is increased by 175°F; both are dangerously over the prescribed limit. Comparing cases I and J indicates clearly the benefit of lowering the compression ratio. The reduction of 1,250 psi in mechanical load (as represented by Pmax) is dramatic.

The computer engine data show the following:

- It is feasible to reach the 350 to 400 BMEP level by selecting the best cycle option without sacrificing unduly the operating life (i.e., reasonable Pmax and Twg levels).

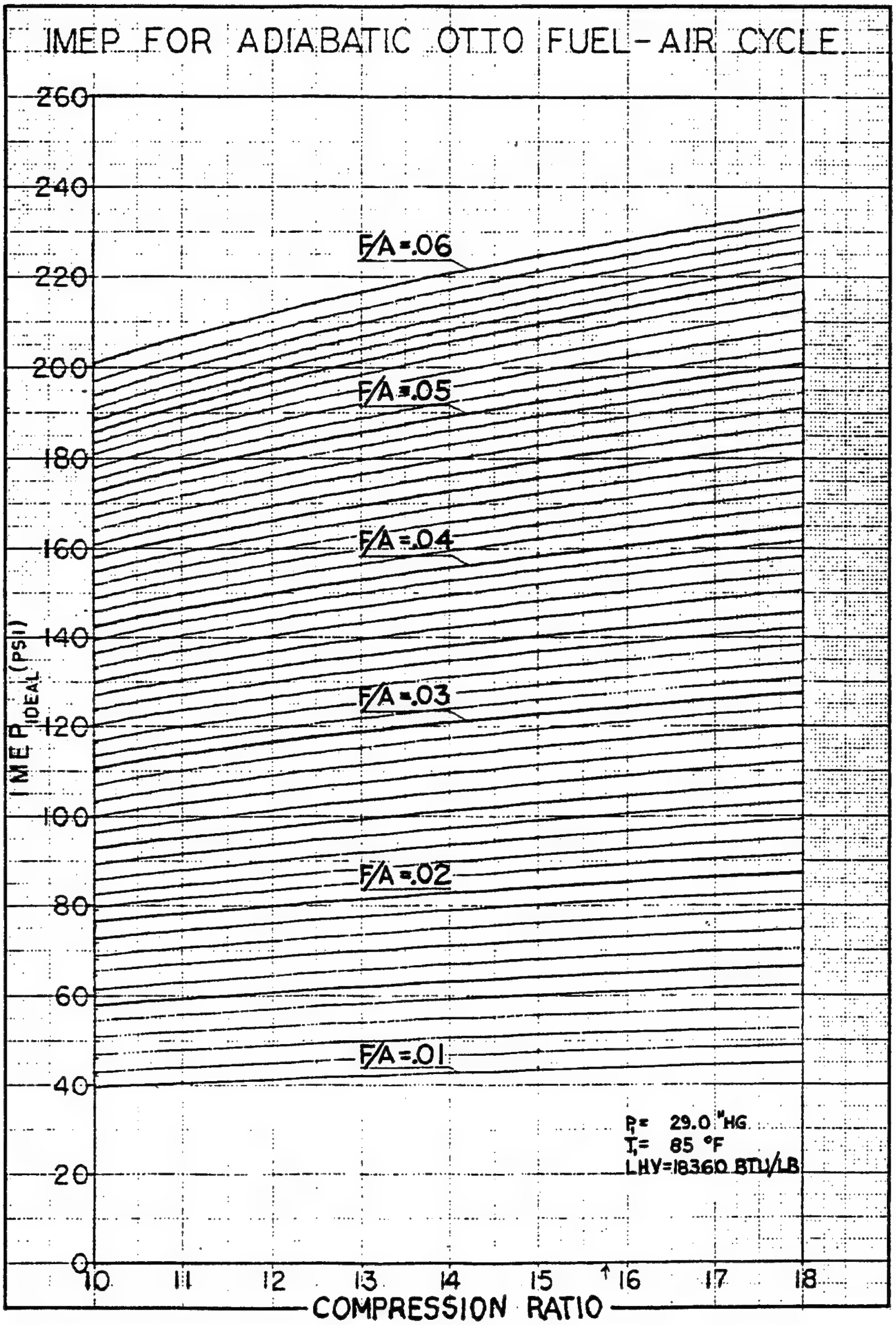


Figure IX-8. IMEP for adiabatic Otto *F/A* cycle

TABLE IX-8. Engine Data

Engine	Bore	hp	Weight/hp	hp/cu. ft	Piston Speed, ft/min
Commercial Engines					
Cat 3208	4.5	210	6.31	9.04	2,333
DD 6V-53T	3.87	225	7.53	4.76	2,100
DD 6-71TAC	4.25	270	8.13	5.01	1,750
DD6V-92TA	4.84	335	6.03	8.58	1,750
DD6V-92TTAC	4.84	270	7.48	6.73	1,625
Cummins L-10	4.921	240	7.08	6.8	2,142
Cat 3408	5.4	450	7.22	6.75	2,100
Mack EM9-400R	5.38	400	7.27	8.29	1,558
Cummins NTC-400	5.5	400	6.48	6.83	2,100
Cummins KTA-2300	6.25	1,200	6.82	7.94	2,188
Military Engines					
DD 6V-53T	3.87	275	4.91	8.09	2,100
DD 6V-92TA	4.84	550	3.67	14.01	1,916
DD 8V-TAC	4.84	736	3.06	14.06	2,083
CV-12	5.315	1,500	3	18.13	2,394
AVCR 1360	5.375	1,500	3.03	12.88	2,167
VTA-903	5.5	600	4.17	15	2,058
Poyaud Hyper	5.591	1,500	3.16	30.16	2,133
MTU 883	5.67	1,474	2.47	30.9	2,756
AVDS 1790	5.75	1,200	4.08	8.98	2,300
dAIPS	5.91	1,450	2.88	31.63	2,218
MTU 873	6.693	1,474	3.87	11.7	2,990
Other					
Dodge V-10	4	400	1.79	27.72	3,233
Arctic Cat	2.52	163	0.92	69.04	3,248

- A low compression ratio operating at high load is vital to reducing the mechanical load (indicated by Pmax). This could be achieved either by variable compression ratio (VCR) or by a variable intake timing (VIT) configuration.

- Aggressive aftercooling (e.g., air-to-air) is necessary for keeping Twg (or Trr) at reasonable levels until a range of high-temperature, power-cylinder materials and lubricants are successfully demonstrated at high fuel-air ratio, high BMEP conditions.

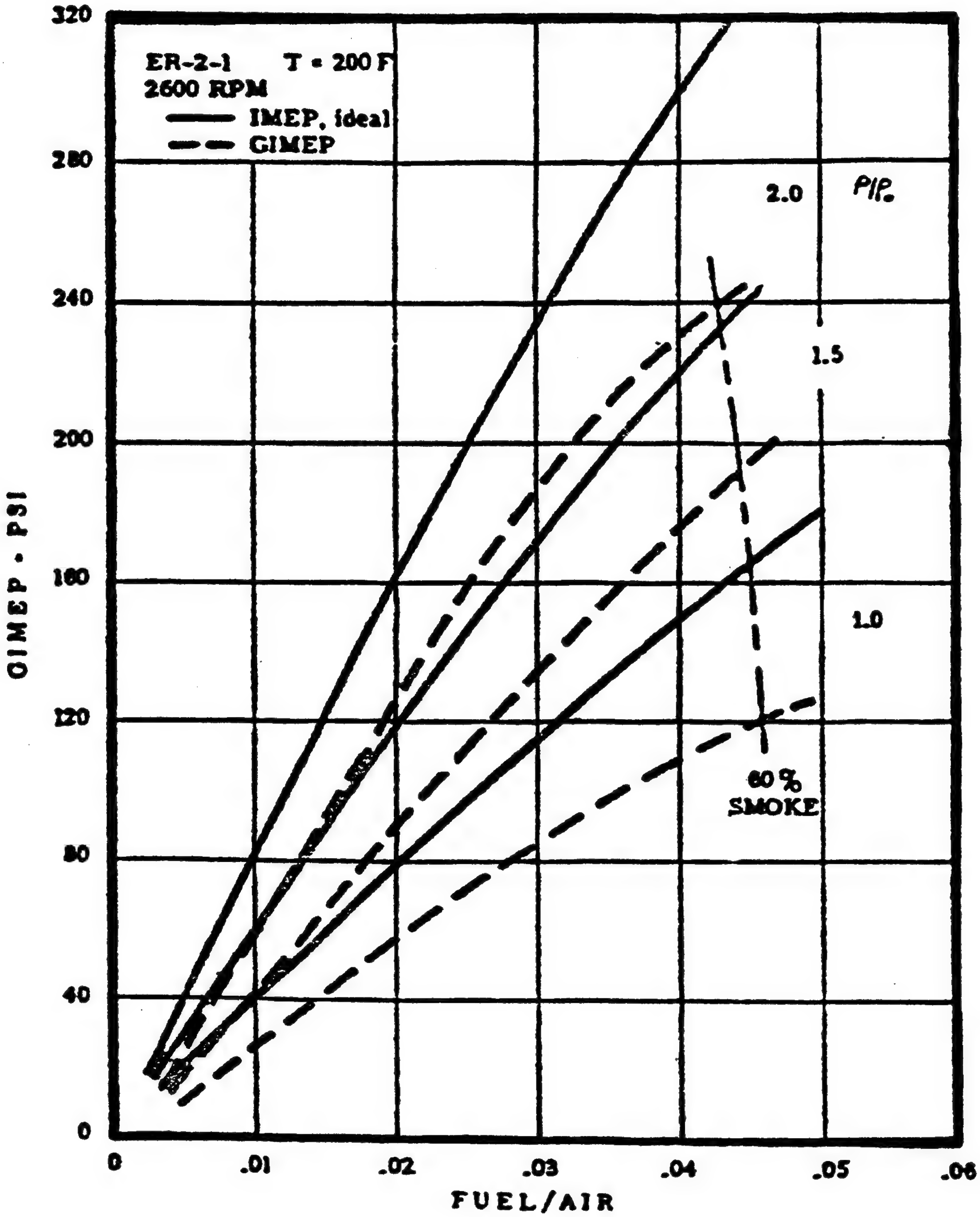


Figure IX-9. IMEP vs. *F/A*, experimental data

TABLE IX-9. High BMEP Without Reducing the Compression Ratio and/or Without Aftercooling

Case	Density Ratio	F/A	After Cooling	Pmax	Twg	CR	BMEP
A	2.0	0.033	0.50	2,700	445	16	190
G	2.0	0.033	0	3,200	480	16	175
H	2.0	0.050	0	4,100	620	16	265
I	2.0	0.050	0.90	3,000	530	16	290
J	2.0	0.050	0.90	1,750	520	10	280

- Running at high BMEP (300 to 400 range) could be achieved through running a rich mixture (such as 0.050 *F/A*) or running at a high inlet density ratio with a moderately lean mixture (such as 0.033). The disadvantage of running a rich mixture (0.050 *F/A*) is the high gas side wall temperature (of an increase of 100°F or more from *F/A* = 0.033) as shown in Fig. IX-10. Aggressive R&D in local cooling is required to keep the critical gas side wall temperature under control.

d. Engine efficiency and compression ratio

As shown previously, BMEP is directly affected by engine efficiency. BMEP is proportional to efficiency and inversely proportional to BSFC. This section will discuss the importance of maintaining high efficiency at the high BMEP operating conditions (see Ref. 1).

- $$\sum Ee = \textit{Product of Engine Efficiencies}$$

$$\sum E = Eisen \cdot Eadia \cdot Epump \cdot Ecomb \cdot Emech \qquad \text{(Eq. 10)}$$

Eisen = $1-(\frac{1}{CR})^{K-1}$, isentropic cold air cycle efficiency.

Eadia = adiabatic cycle efficiency considering real gas loss. See Fig. IX-11, which is taken from Ref. 5.

Epump = pumping efficiency, affected by gas exchange flow pumping losses and energy recovery performance by turbocharger.

Ecomb = combustion efficiency, affected by *F/A*, injection performance, mixing and rpm. Efficiency drops off drastically when the stoichiometric ratio is approached even at naturally aspirated conditions.

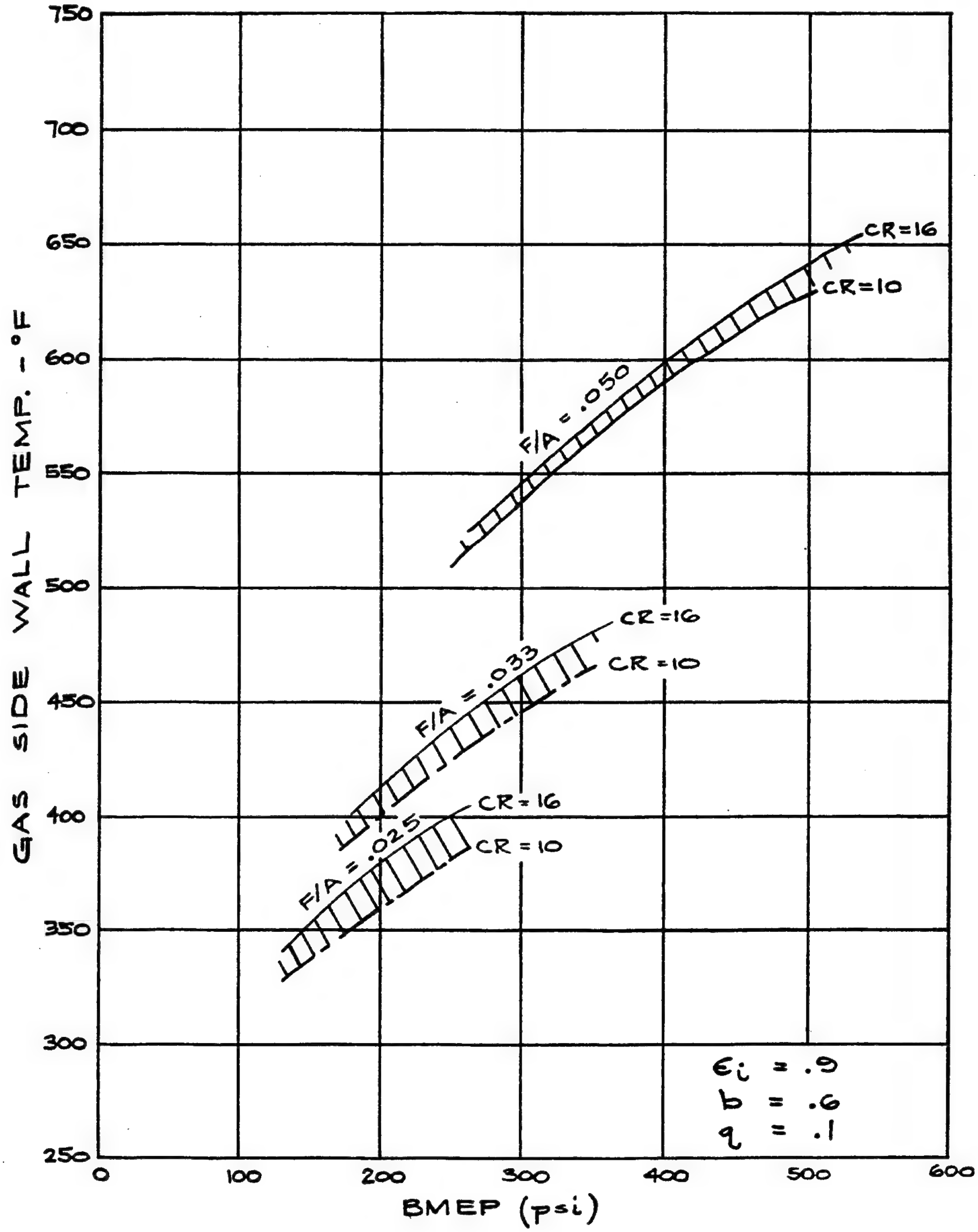


Figure IX-10. <u>BMEP vs. gas side wall temperature</u>

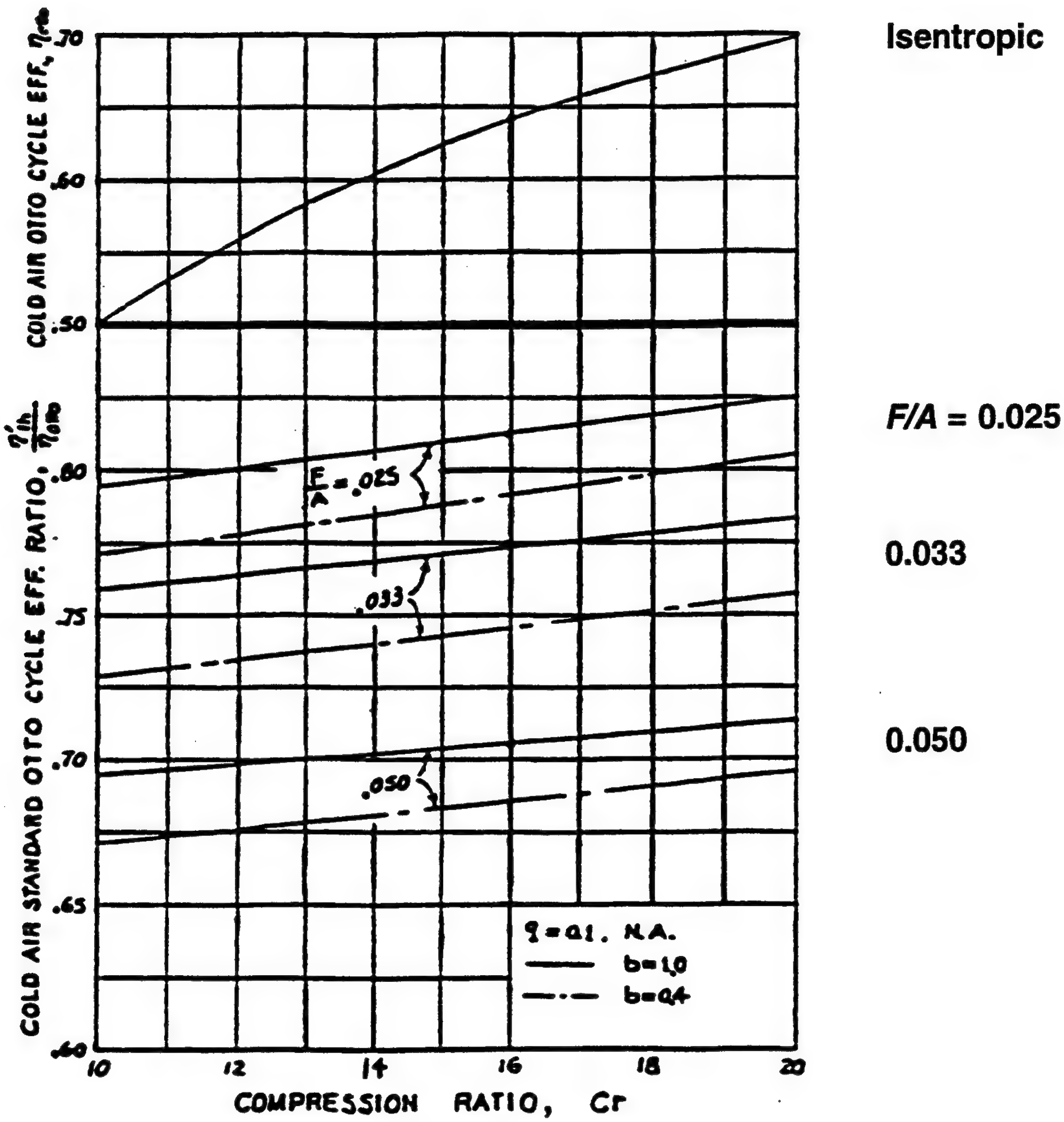


Figure IX-11. <u>Effect of compression ratio and fuel-air ratio on indicated cycle efficiency</u>

Emech = mechanical and accessory efficiency affected by cylinder pressure, piston speed, design sophistication, tribology, and accessory load required.

e. Compression ratio and variable inlet valve timing cycle

For most vehicular diesels, the compression ratio is kept at a fairly high level (approximately 15 to 16) for cold starting and fuel efficiency. The isentropic efficiency is a function of the expansion ratio (or the compression ratio, given that they are the same for most engines). This is shown in Fig. IX-12. The exhaust loss is represented by $(1/CR)^{K-1}$ and is not recovered for naturally aspirated engines. The merit of a variable compression ratio engine was studied and verified by the Army in the 1960s and 1970s. Additional R&D could be conducted to develop a practical and reliable variable configuration. A variable effective compression ratio cycle with variable inlet valve closing timing is a recent development that has not yet been studied in conjunction with a diesel engine. The advantage of this variable inlet timing (VIT) cycle is as follows:

- Improvement of isentropic cycle efficiency – The expansion ratio is larger than compression ratio at running conditions. Compression PV work is reduced by late or very early inlet valve closing.

- Cold starting capability can be enhanced by maintaining a high compression ratio at starting conditions. The inlet valve closing will be close to bottom dead center.

- The end gas compression temperature and pressure are lowered substantially at high BMEP conditions, thereby reducing cylinder pressure and cycle temperature substantially. This will reduce the mechanical and thermal loads. A long expansion ratio would reduce the exhaust temperature, which also reduces the average wall temperature and thereby thermal load.

- Depending on wall temperature, early intake valve closing before bottom dead center (called Miller cycle) could have the added advantage of some internal cooling during the piston expansion, thereby reducing thermal loads. This would not be effective for a low heat rejection engine where the average wall temperature is higher than the engine inlet temperature.

This leads to the conclusion that a high isentropic efficiency, achieved through a high expansion ratio (16 to 18) and a flexible compression ratio (9 to 16) configuration, can be accomplished through the use of a VIT mechanism. Further development of this mechanism should therefore be included in a high power density engine R&D program.

f. Adiabatic efficiency

Adiabatic (real gas loss) efficiency is defined as the efficiency ratio of the real gas adiabatic cycle (no loss) and that of the cold air constant volume, isentropic cycle. The gas exponent n of the adiabatic cycle is dependent on the *F/A* ratio and temperature, while the cold air exponent of the isentropic cycle is $k = 1.4$.

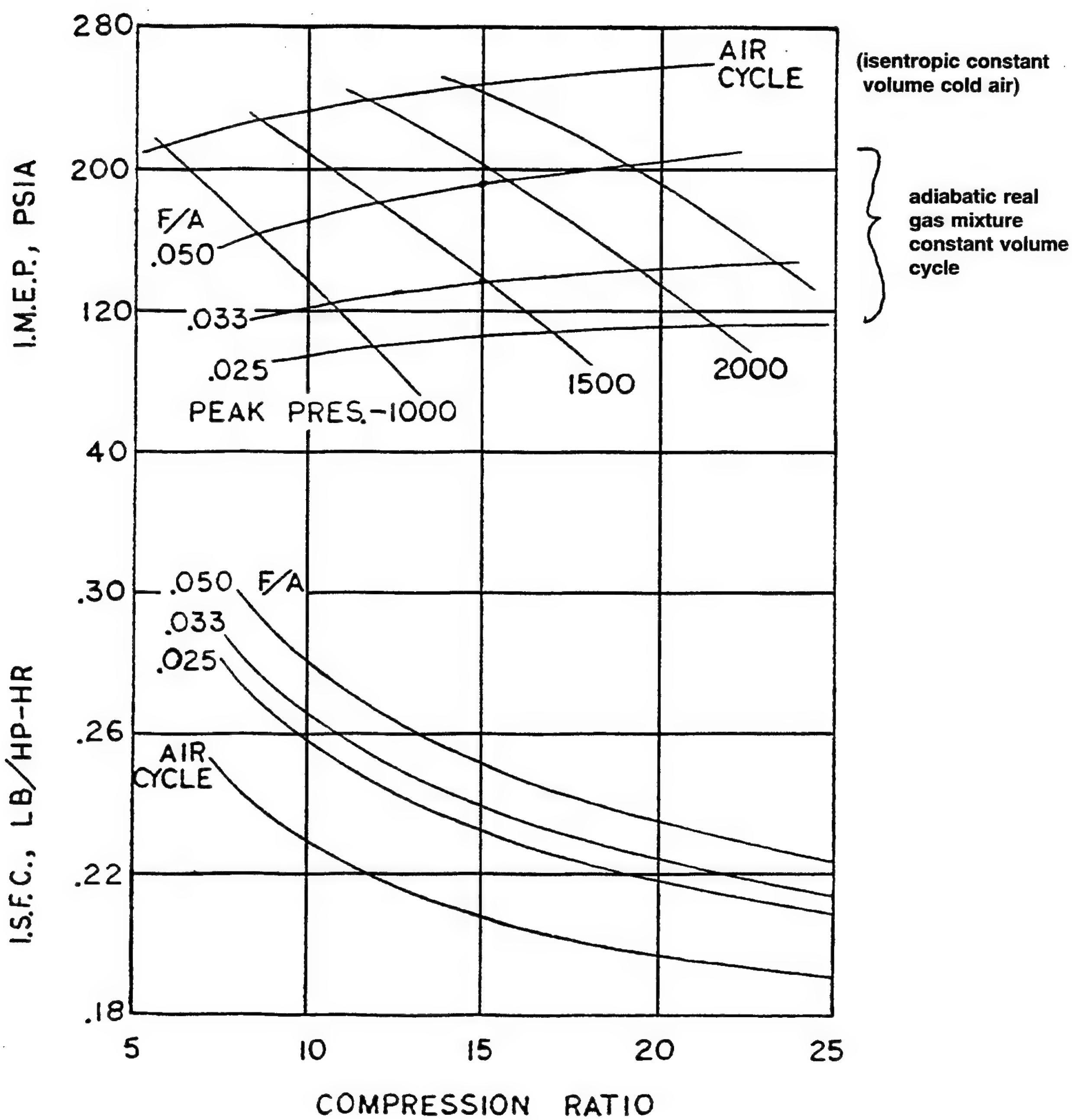


Figure IX-12. <u>Constant volume cycles</u>

The difference between the isentropic and adiabatic efficiency is in real gas loss. The adiabatic (no heat loss) gas exponent of a fuel-air mixture is lower than that of cold air (k = 1.4). For a computer engine with CR = 16, B = 1.0 (constant volume combustion), and F/A = 0.05, the computed adiabatic efficiency is 71.5 percent. The calculated isentropic efficiency (cold air cycle) is 64 percent with 36 percent exhaust loss. This calculation is shown in Fig. IX-12. The combined indicated real gas no heat loss cycle efficiency ($Eisen \bullet Eadia = Ei$) is 45 percent. For lean burning with a fuel-air mixture of 0.033, the adiabatic efficiency is 77.4 percent, a 4.5 percent improvement over the rich burn cycle with a F/A of 0.05. This substantive 4.5 percent efficiency difference is due to real gas burning and expansion at a moderately rich mixture (F/A = 0.05), as compared to an ideal cold air cycle with k = 1.4.

Running at a rich mixture ratio will hurt fuel economy regardless of combustion efficiency. For military engines, however, power density rather than fuel economy is the primary goal. Operating at a moderately rich F/A ratio could be tolerable as long as the smoke emitted is not visible (which would draw enemy fire). The smoke limit (12 percent opacity) may very well be the factor determining the rich mixture limit for military diesels. The adiabatic loss due to real gas burning can hardly be avoided.

g. Combustion efficiency

Combustion efficiency is determined by comparing the actual compression expansion indicated PV card (IMEP) with the theoretical constant volume indicated PV card considering real gas. The combustion loss defined here includes heat loss during combustion, late combustion, mixing, incomplete combustion, etc. Combustion deteriorates quite noticeably when the smoke limit is exceeded. An experimental combustion efficiency curve plotted versus F/A is shown in Fig. IX-13.

These engines were tested at NA conditions. For turbocharged engines, the combustion air is more dense and the smoke limited F/A ratio would be progressively reduced (see Fig. IX-9). Combustion loss at rich F/A ratios is a major research problem for heterogenous combustion, especially at high inlet density ratio conditions.

Extensive research on injection, mixing, and combustion will be required to reduce combustion loss at high F/A (0.040 to 0.050 range) conditions. High injection pressure and flexible injection timing should help in boosting combustion efficiency by providing desirable injection rates for a wide range of operating conditions. Extensive combustion and injection research will be needed on burning rich mixture (in the range of 0.040 to 0.045) at highly turbocharged conditions.

Combustion efficiency test data are included in Fig. IX-13. Plotted against the fuel-air ratio, the test efficiency drops sharply when the F/A ratio is increased beyond 0.040 for naturally aspirated engines. These well-developed engines were smoke limited (68 percent smoke in 1963) and just short of 0.06 F/A (Ref. 5). In Fig. IX-13, Engine A is a development lab engine. Engine B is a production engine of the same overall combustion design, except that the production tolerance, etc., reduces the air utilization factor by almost 10 percent. The burning of oxygenated fuel could be another approach to combustion loss reduction in the high fuel-air ratio condition.

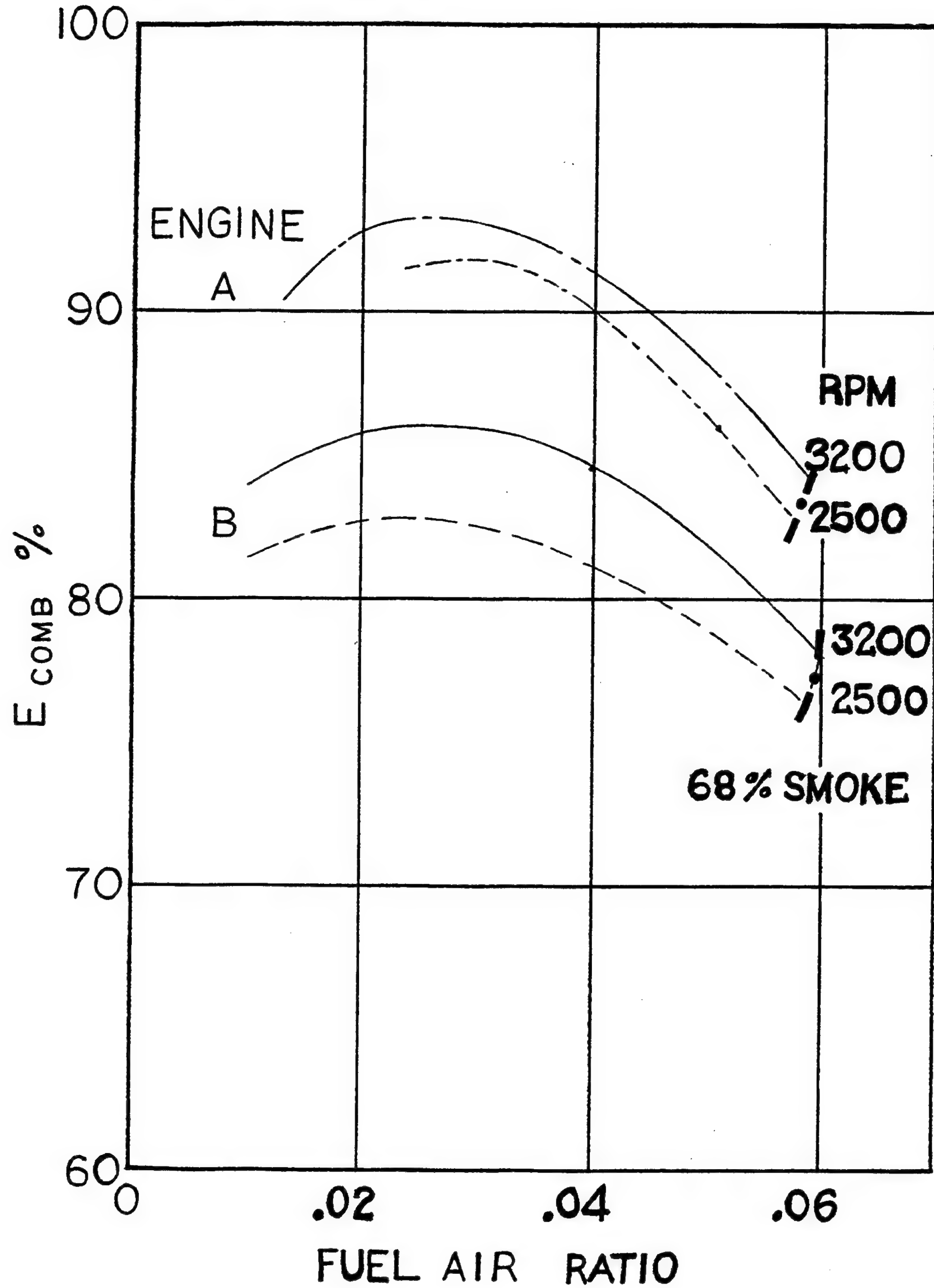


Figure IX-13. <u>**Combustion efficiency, naturally aspirated conditions**</u>

h. Pumping efficiency

Pumping efficiency (*Epump*) is determined by evaluating the intake and exhaust stroke of the indicated card. Pumping efficiency is defined as follows:

- $$Epump = (1 - \frac{PMEP}{IMEP_n})\% \qquad \text{(Eq. 11)}$$

- $$IMEP_n = net\ IMEP = IMEP \pm PMEP \qquad \text{(Eq. 12)}$$

PMEP = Pumping mean effective pressure

IMEP = Indicated mean effective pressure.

For a naturally aspirated engine, the pumping loss is the gas exchange loss (or inlet-exhaust valve flow loss, i.e., a negative pumping loop - see Ref. 1, p. 6). This negative pumping loop could be significant at high piston speeds. For a turbocharged engine, some small percentage of exhaust energy (and therefore, fuel input) is recovered in the expansion of the compressed inlet air in the intake stroke. This favorable condition exists when the inlet air is compressed to an inlet pressure level that is higher than the exhaust back pressure. This is possible when the combined turbo efficiency is in excess of 55 percent. A positive pumping loop is formed (ΔPie is positive rather than negative). This "pumping gain" results in a pumping efficiency larger than 100 percent, as shown in Equation 11. The pumping efficiency, thus defined, provides a measurement on how much exhaust energy (as a fraction of fuel energy) is recovered. The actual exhaust recovery is somewhat higher than the Δpsi *Epump* indication, as the turbocharging also overcomes the gas exchange loss or the negative pump loop loss (when the turbocharger is not installed). This discussion further emphasizes the importance of having low flow loss through the valve and manifolding and having advanced turbocharger efficiency for high BMEP, high rpm cycle development. The generalized pumping loss equation (Ref. 1) is as follows:

- $$PMEP = K \cdot \frac{\rho}{\rho_0} CM^2 + \Delta Pie \qquad \text{(Eq. 13)}$$

ρ/ρ_o = density ratio

CM = fpm (piston speed)

ΔPie = inlet pressure minus exhaust pressure.

This correlation is shown in Fig. IX-14.

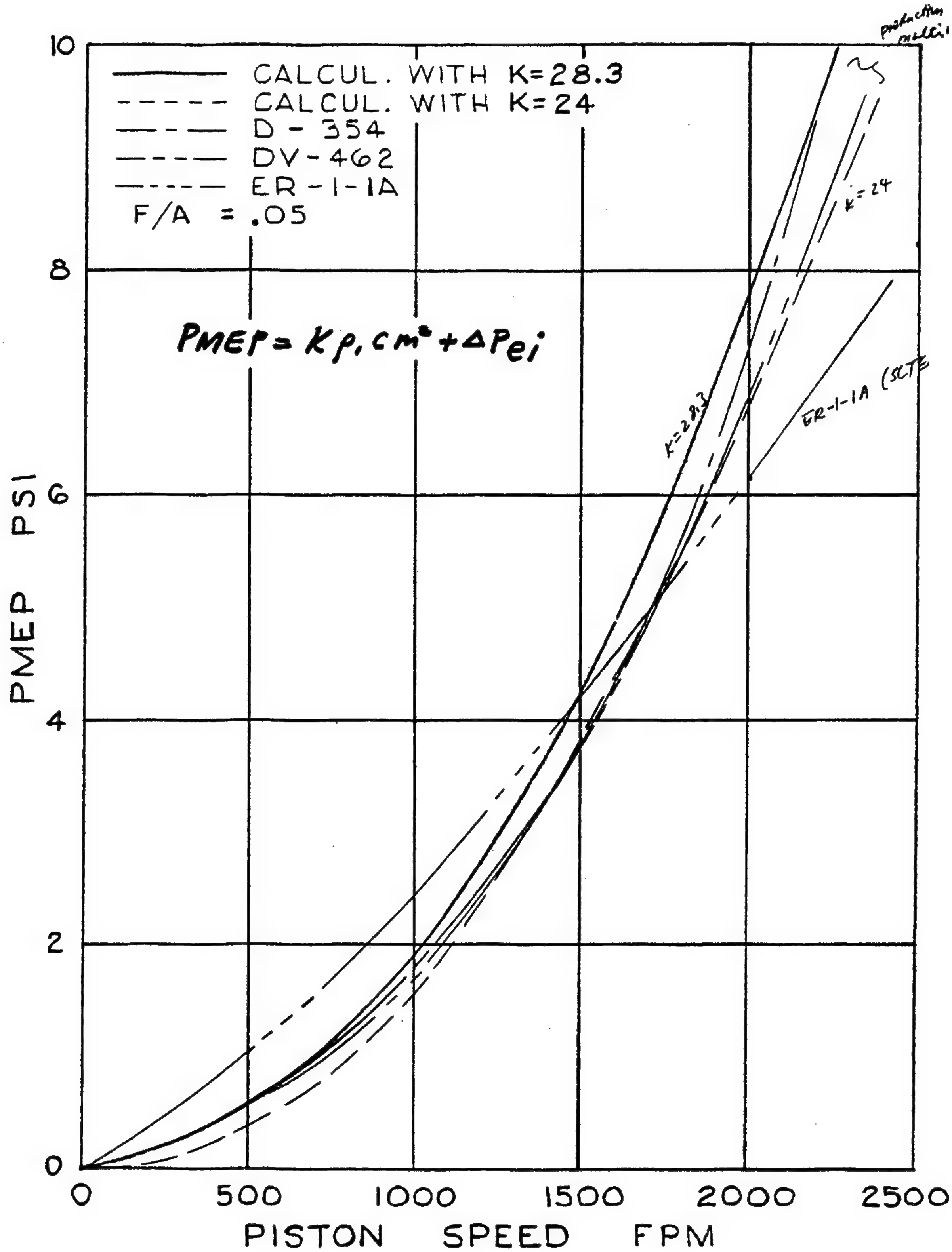


Figure IX-14. <u>Pumping loss calculation</u>

In reviewing the competitive engine data, the Poyaud Hyperbar engine appears to have a superior turbocharging efficiency. Its pressure ratio exceeds 6:1, and a PMEP gain of 30 psi (P_2 - P_3) is realized (see Ref. 4, p. 42). Pumping efficiency is calculated per Equation 11 to be 106 percent at the rated condition. From this observation, it can be concluded that the best possible turbocharger configuration should be sought for TACOM's high performance engine. A 6 percent gain in fuel efficiency and BMEP is a major improvement and should not be overlooked.

i. Mechanical efficiency

Mechanical efficiency provides an evaluation of the mechanical loss, including accessory work and rubbing frictions. The loss can be measured by the difference of the net indicated work (IMEP - PMEP) and the brake output (BMEP).

The shaft horsepower required to drive the cooling fan, pumps, injection pump, etc. is not trivial. It should be minimized by careful matching and flow controls. Fan horsepower for the heat exchanger in particular should be scrutinized.

Rubbing loss is somewhat more difficult to analyze. It covers piston and ring friction, bearing loss, valve train friction gear loss, seal friction, etc. A rapid increase in rubbing loss is an indication of rapidly deteriorating tribology conditions and will result initially in high wear and ultimately in scuffing and seizure.

The rubbing and accessory mean effective pressure (RAMEP) correlation shown in Ref. 2, p. 11, is as follows:

- $$RAMEP = A + B \cdot CM + C \cdot Pmax \qquad \text{(Eq. 14)}$$

This equation is correlated by running many indicated PV diagrams on well developed production engines. The variables include piston speed (Cm or fpm) and Pmax. The RAMEP correlation for the ATAC single cylinder engine is reproduced in Fig. IX-15. The rubbing loss could deteriorate when Pmax is increased (due to deformation, reduced clearance, etc.). A variable compression ratio condition should improve mechanical efficiency by controlling Pmax and rubbing loss.

Mechanical efficiency improvement will stem from the following design and component developments:

- meticulous matching of the accessory load with the engine needs (cooling fan, oil pump, water pump, etc.);
- detailed refinement of the piston, sleeve, crankshaft, and bearing design for maximum rigidity and minimal thermal deformation at the high BMEP operating conditions;

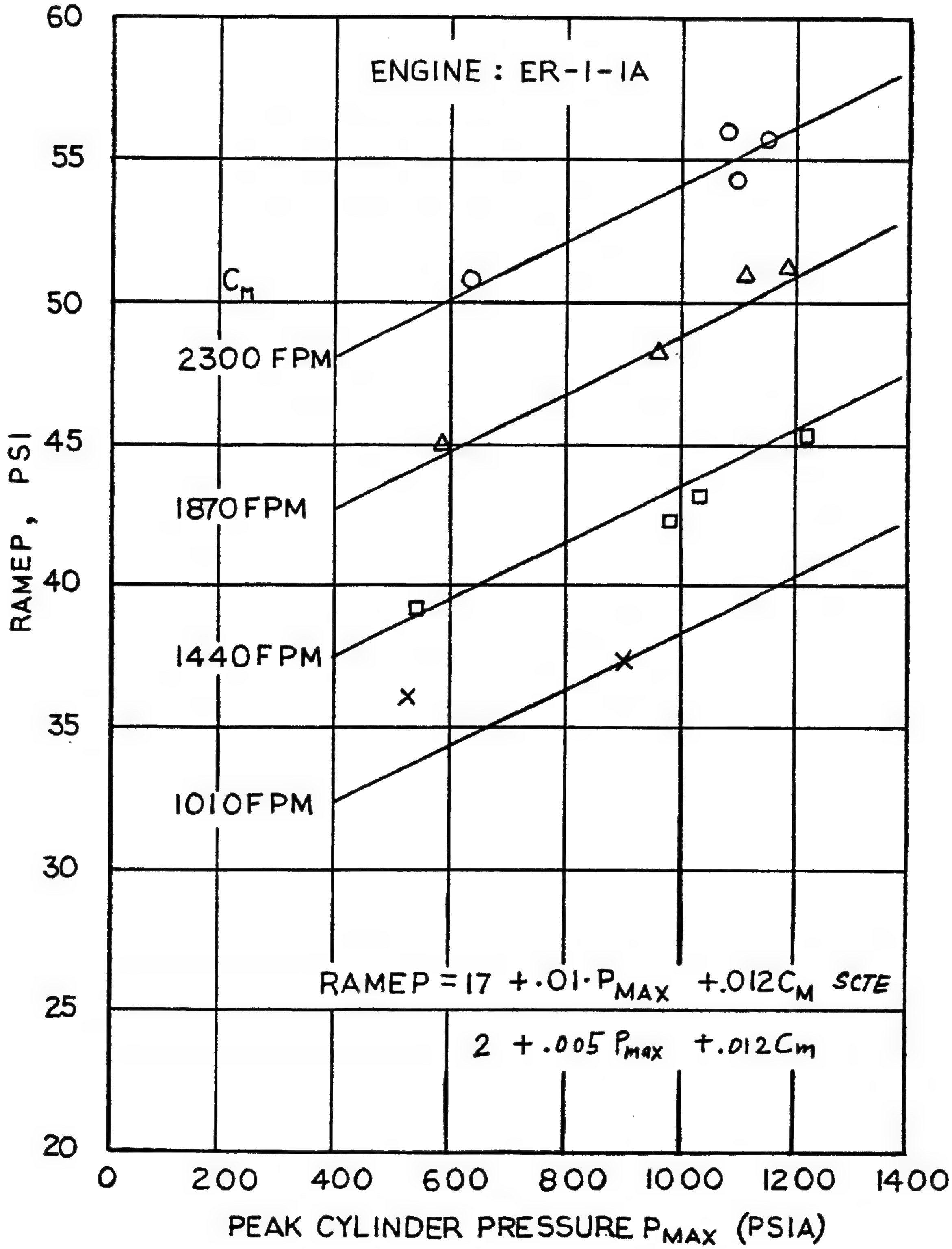


Figure IX-15. RAMEP vs. peak cylinder pressure

- high-temperature lubrication and material for the piston, ring, and sleeve interface;
- light inertia weight of the reciprocating and rotating components to reduce bearing load and imbalance vibration at high rotating speeds.

Selection of a cycle option which will not cause high mechanical load and poor tribological conditions at the predicted operating temperature and pressure is recommended.

The engine efficiency study presented here provides a step-by-step analysis on how the fuel energy is converted to brake work. The purpose of the study is to select a cycle (and cycle parameters) which will provide

- minimal exhaust loss achieved through a high expansion ratio, resulting in high isentropic efficiency;
- minimal real gas loss, achieved through lean burning (not too rich), resulting in high adiabatic efficiency;
- minimal combustion and heat transfer loss, achieved through high fuel injection pressure, resulting in high combustion efficiency;
- minimal gas exchange loss and maximum turbo exhaust energy recovery achieved through high turbo efficiency, a 4-valve layout, and streamlined manifolding and porting, resulting in high pumping efficiency;
- minimal accessory load and rubbing friction achieved through advances in mechanical design and tribology, resulting in high mechanical efficiency.

A combination of variable inlet valve timing, moderately rich burning, aftercooling and local cooling, and a high density ratio engine configuration would provide the synergy needed to achieve both high power density and a reasonable fatigue life for future 350 to 400 BMEP intermittent combustion engines. When this level of BMEP is coupled with an engine piston speed of 2,700 ft/min, the future dAIPS engine could be rated at 2,500 hp with a power density of approximately 48 hp/cu. ft, a 50 percent improvement over today's dAIPS engine.

IX.2.2.3 Summary and suggestions

a. Summary

A. Computer analysis could help to achieve the high power density goals through selection of the most suitable cycle configuration and cycle parameters, and by avoiding those cycles which would lead to unacceptable mechanical, thermal, and combustion conditions.

B. A high power density (and high BMEP) engine 50 percent beyond the dAIPS engine level is feasible through development of a variable inlet valve timing configuration in conjunction with aggressive development in turbocharging, aftercooling, combustion and injection, high temperature materials, and lubricants.

b. Results and suggestions

A. Cycle Simulation – Develop a high BMEP, high performance cycle analysis for the purpose of evaluating innovative cycle approaches in high power density engine systems. The cycle should be capable of predicting cylinder pressure and cycle temperature for the purpose of estimating mechanical load, thermal stress, and the thermodynamic losses. This could be an in-house TACOM program.

B. Variable Intake Valve Technology – Take advantage of a variable inlet valve timing cycle to reach the desired high BMEP levels without exceeding prescribed mechanical load limit of the engine. The objective is to have a high expansion ratio configuration together with a low compression ratio except for starting and idling. This should be a long range R&D program.

C. Combustion Development – Increasing the fuel-air ratio by approximately 15 to 20 percent from its present level of 0.033 (30:1) should be a development priority to provide the desired 20 to 30 percent emergency combat power boost beyond the rating. This is a combustion research program.

D. Cooling Optimization – Provide aggressive aftercooling (with a small penalty in envelope increase) to develop effective local cooling and to counteract the thermal load increase caused by burning rich mixture (which is desirable for compactness) and increasing power density.

E. Turbocharger Enhancement – Develop efficient high pressure ratio turbochargers (one or two stage) to reduce the amount of aftercooling required and to increase the pumping efficiency through aggressive exhaust heat recovery.

F. Increasing Displacement and Air Utilization – Use a combination of high piston speed (up to 2,800 ft/min), high density ratio (30 percent), and high *F/A* ratios (0.038) to attain the goal of improving the power density ratio by 50 percent or more.

G. Single Cylinder Test Engine – Conduct research on material, tribology, and power cylinder thermal load capability to accelerate the single cylinder test engine research and development program for determination of material and stress limits.

- High piston speed (up to 2,800 fpm level);
- High BMEP (up to 400 psi level);
- High strength, low weight material, components for high mechanical and thermal load limit extension;
- Variable inlet timing (VIT) mechanism.
- New high temperature synthetic lubrication.

H. Cooperative R&D With Industry – A research and development program could be conducted under the general umbrella of "Defense Conversion," given that a great deal of knowledge has been accumulated at TACOM and Army Research in system integration and the development of low heat rejection components. These same concepts would be beneficial for commercial applications, including fast boats, emergency gensets, diesel locomotives, heavy construction equipment, etc. The commercial engine could be a long life derated version of the military engine. With enthusiastic industry support, this new research and development program would entail very little risk.

I. Engine Family Development – Vehicle and hp requirements for each potential military application cannot be projected with any certainty 10 to 20 years into the future. A wise approach would include flexible planning to cover a wide hp range, with design based on modularity concepts suitable for agile machining. This could result in an engine family of approximately 100 hp per cylinder with a range in ratings from 600 hp (V-6) to 1,600 hp (V-16). The engine family might well have a "dual use," with the super high power density, short-stroke version used by the Armed Forces and a commercial, long-stroke version for various high performance vehicular, power generation, and marine applications.

IX.2.2.4 List of reference works

1. "A Review of Engine Advanced Cycle and Rankine Bottoming Cycle and Their Loss Evaluations." Chen, Simon K. and Lin, Rocky, SAE Paper No. 830124.

2. "Development of a Single Cylinder Compression Ignition Research Engine." Chen, Simon K. and Flynn, Patrick F., SAE Paper No. 650733.

3. "High BMEP Cycle Analysis." Chen, Simon K. and Wu, Tang, IHER Memorandum Report No. 66, Project No. 2-BH.

4. "Foreign Diesel NATO Comparative Test Program – Unidiesel V8X 1500 Engine Evaluation." Lemmo, Alfred C., TACOM RD&E Technical Report No. 13537.

5. "Development and Evaluation of the Simulation of the Compression-Ignition Engine." McAulay, K.J., Wu, Tang, Chen, Simon K., Borman, G.L., Myers, P.S., and Uyehara, O.A., SAE Paper No. 650451.

IX.2.3 Steady-flow machinery

IX.2.3.1 Turbocharger

Figure IX-16 shows the power of the dAIPS, MTU 883, and Perkins CV12 engines compared with other engines in the SwRI engine design data base. It is interesting to see from this plot that these engines, although they are among the best in terms of power/unit displacement, do not stand out significantly in this respect.

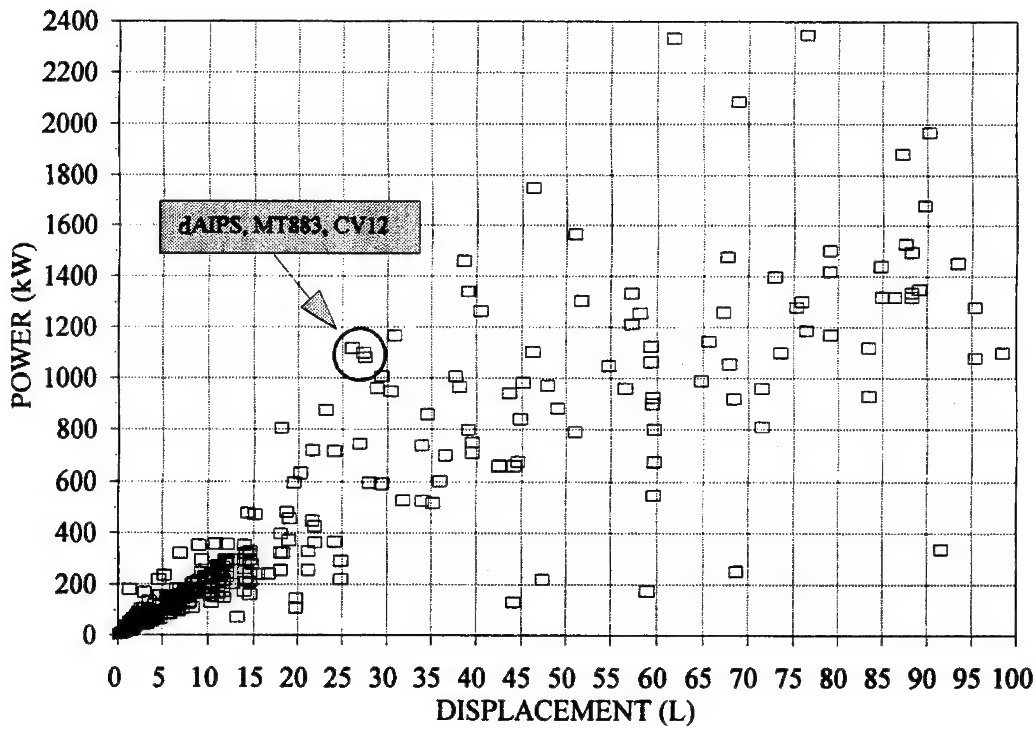


Figure IX-16. <u>SwRI engine design database (survey of CI engines < 100 liters)</u>

Increasing the density of the air supplied to a given diesel engine is crucial to increasing the power from that engine. For constant air/fuel ratio, specific power is directly proportional to intake manifold density. There are two methods of increasing intake manifold density: 1) increasing the boost pressure or 2) increasing the degree of aftercooling. Until recently, compressor pressure ratios have been limited to approximately 3:1. To achieve increased boost pressures, it has been possible to use two-stage turbocharging. Two-stage turbocharging is defined as two turbochargers functioning in series; the combined efficiency is the average of the individual turbocharger efficiencies. The disadvantage is that two-stage turbochargers have a narrow combined map width and are difficult to package.

There have been recent improvements in turbocharger technology that have increased compressor pressure ratios so that pressure ratios of 6:1 are now possible from a single-stage compressor but with reduced map width. However, it is important to maintain compressor map width for acceptable engine performance at peak torque as well as at high speed. Additional research and development is required to increase the compressor pressure ratio while maintaining or widening the map width. Increasing the speed capability of the compressor and turbocharger wheels would permit greater pressure ratio without reducing map width. But compressor speed is limited by the strength of the wheel material. Wheel tip speed is a measure of the centrifugal forces in the wheel, and compressor pressure ratio is approximately proportional to tip speed for a given wheel. Current commercial aluminum compressor wheels have tip speeds up to 600 m/s. The low grade titanium compressor wheel made for the Diesel Engine Component Technology (DECT) program demonstrated 625 m/s tip speed. Slightly higher tip speed could be achieved with magnesium or high grade titanium. Steel wheels could also provide high tip speeds, but these would have unacceptable inertial response. Ceramic compressor and turbine wheels would

allow tip speeds of over 650 m/s, but the technology of bonding these ceramic wheels to the turbocharger shaft requires additional development.

Figure IX-17 shows a plot of turbine intake temperature against air/fuel ratio for a modern on-highway heavy-duty diesel engine. It is clear that if combat vehicle engines are to operate closer to stoichiometry, the exhaust temperatures will significantly increase. This will in turn decrease the strength of conventional turbine wheels and so, limit pressure ratio. The strength of ceramic turbine wheels is not affected by temperature, so they have an additional advantage in this respect. It would appear the greatest gains in turbocharger pressure ratio, while maintaining map width for least effort, could be achieved by developing the technology of ceramic compressor and turbine wheels. Ceramic turbine wheels are now in production on gasoline automobile engines. Additional areas of development are improved bearing that would be necessary to withstand the increased thrusts from operating at high pressure ratios. Compressor inlet guide vanes could also increase map width.

Figure IX-18 plots the calculated effect of increased compressor efficiency upon compressor out temperature and in turn, its effect on the amount of heat rejected through the aftercooler to the coolant as a fraction of the total heat rejection. It can be seen from this plot that increasing the compressor efficiency does not have a great impact on heat rejection and therefore, may not be justified for this reason alone.

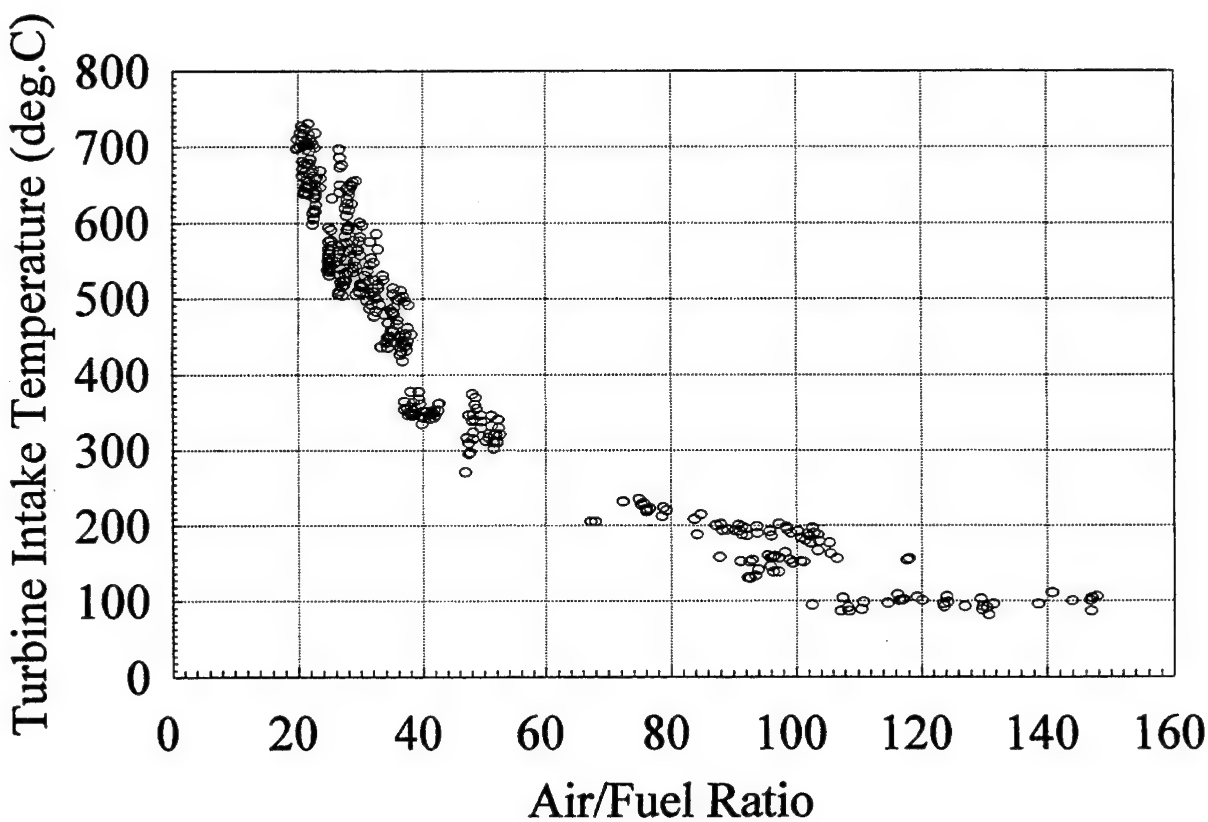


Figure IX-17. <u>Effect of air/fuel ratio on turbine intake temperature</u>

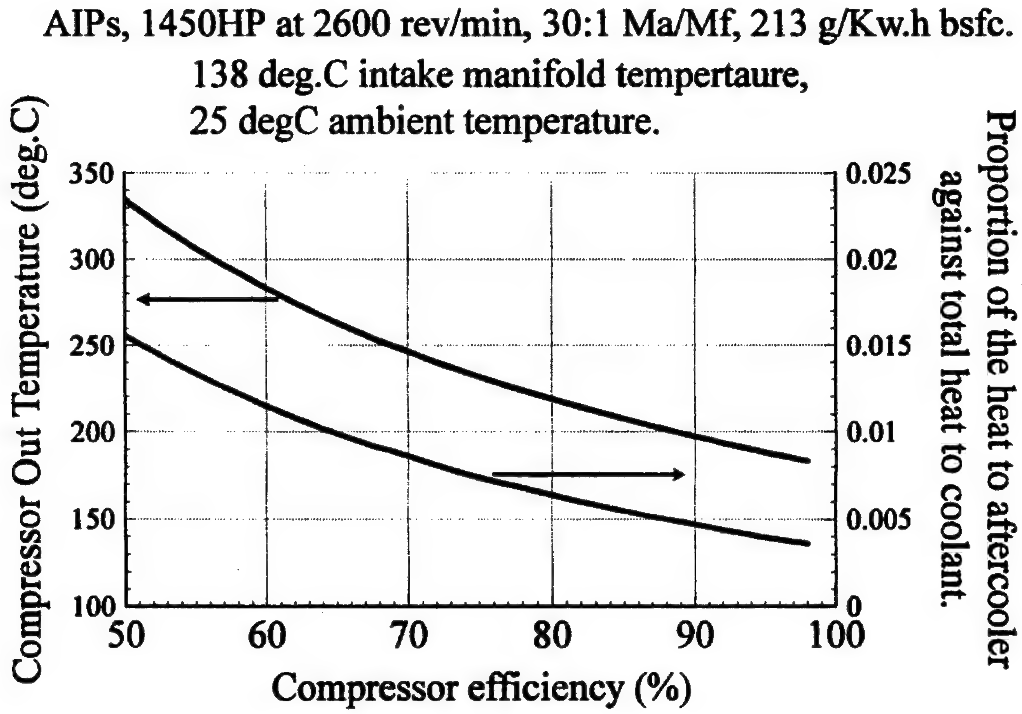


Figure IX-18. **Effect of increased compressor efficiency on heat rejected to the coolant**

The dAIPS propulsion unit has the requirement of accelerating from idle to full power in two seconds. This is a very severe requirement as the turbocharger cannot accelerate fast enough to provide sufficient air to support combustion. The result is excessive smoke during accelerations to the extent that Cummins has been forced to introduce a particulate trap to the dAIPS engine to reduce the black smoke signature. The particulate trap, together with the necessary equipment to regenerate it, takes up significant space. It also increased fuel consumption by adding back pressure to the engine. The particulate trap only obscures the effects of poor engine acceleration; it does not address the basic poor response of the engine. There are more elegant solutions to this problem, and some of these are listed below.

- Decreasing turbocharger inertia will increase air/fuel ratio during acceleration. A ceramic compressor and turbine wheel will achieve this.

- Variable geometry turbochargers and wastegated turbochargers can help by providing more boost at lower speeds, but these alone are unlikely to achieve the required acceleration.

- A supercharger could be placed upstream of the turbocharger. The advantage of a supercharger is that unlike a turbocharger, it responds instantly and also provides boost at low speed. The disadvantage of a supercharger is that it provides too much boost at full power and also causes high fuel consumption. There are easy solutions to these problems. One solution is to use the Vairex supercharger, a recent invention that is undergoing extensive development by the Vairex Company of Boulder, Colorado. This is a compact variable displacement supercharger. The displacement can be maximized at low speed and during accelerations when the turbocharger is ineffective, and turned

to zero when the turbocharger is functioning normally. When the supercharger displacement is set to zero, it is essentially free-running and offers no impediment to the engine power and fuel economy. The Vairex supercharger would require a controller to operate in this way, but this could be an extension of the existing controllers in the propulsion unit. A schematic of this configuration is shown in Fig. IX-19. The supercharger for the application need only measure 10 × 14 × 24 in.

Another method of increasing engine air consumption is to employ pulse tuning into the intake manifold. The basic principle is to size the branches of the intake manifold so that at certain engine speeds, pressure wave action in the manifold passages increases cylinder filling by as much as 20 percent. Helmholz resonators are an extension of this principle where several cylinders share the same resonance chamber. The disadvantage of this for a propulsion unit is that the additional tuning pipes and resonance chambers take up significant space, so this may not be appropriate for this application.

IX.2.3.2 Heat exchangers

The use of high-temperature coolant at 177°C (350°F) that is also the engine lubricant is the most effective way of reducing heat exchanger size. This has already been done on the dAIPS project.

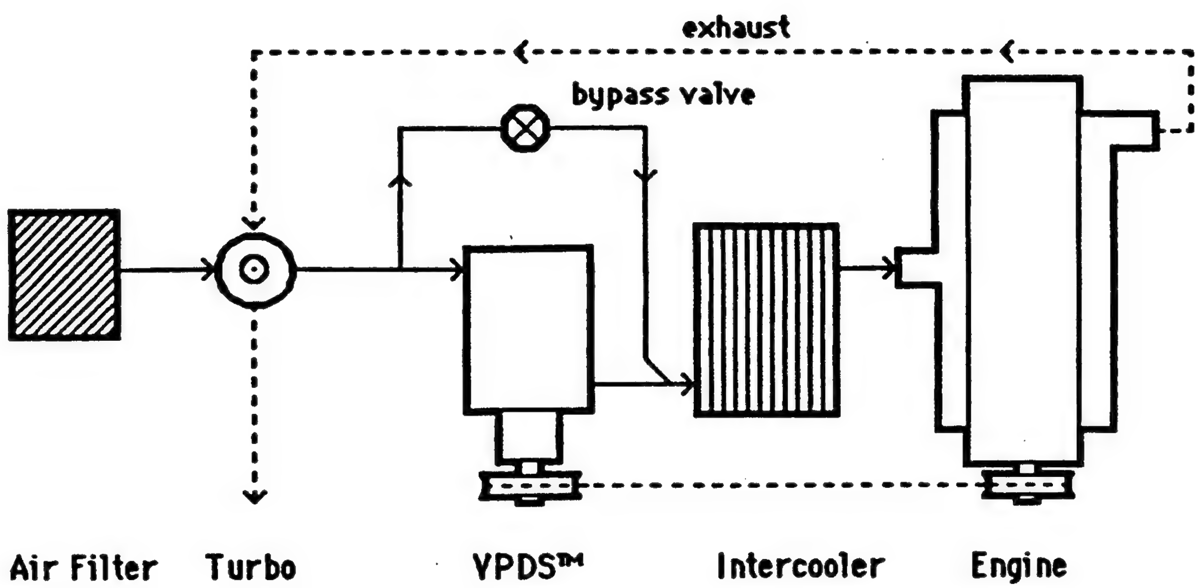


Figure IX-19. Schematic of Vairex supercharger

There appear to be few additional possibilities for improving the basic design of heat exchangers; the design of the turbolizers on the liquid side and the fins on the air side seem well developed.

Approximately 95 percent of the resistance to heat transfer occurs at the air side of the heat exchanger. This resistance increases as the air side surfaces become contaminated with dust so that the dAIPS specifications require the heat exchangers to be effective with 30 percent redundancy. If contamination of the heat exchangers could be avoided, it may be possible to relax this 30 percent redundance requirement, and the heat exchangers could be made proportionally smaller. One possibility is to filter the cooling air by passing it through a set of vortex tubes similar to those used in the engine air supply prefilter. This cooling air filter would be courser than the engine air filter and less restrictive, but if the engine air was drawn from the filtered coolant, it would already be partially filtered and place less demand on the engine air filtration system.

An understanding of the physical mechanism of agglomeration of dust to outside heat exchangers may indicate ways in which it could be inhibited. For example, dust accumulation could be less in creating flow velocity regimes or conditions of boundary layer turbulence. This type of fundamental research could be carried out inexpensively at a university.

As was mentioned earlier, the engine power is proportional to the density of the charge in the intake manifold. Since the only coolant in the dAIPS engine is also the lubricant, the intake air is only cooled from 230°C to 138°C. Intake manifold air temperatures of 40°C are common on heavy-duty on-highway engines with air-to-air aftercoolers. Although it is not expected that 40°C would be achievable with air-to-air cooling in combat vehicle propulsion units, an air-to-air aftercooler would provide significantly lower intake manifold temperatures than are currently achievable with cooling by high-temperature lubricant. Moreover, this may provide a reduction on total heat exchanger volume since with aftercooling by lubricant, the heat from the intake charge is transferred in two stages: first to the lubricant, then from the lubricant to the cooling air. With air-to-air aftercooling, there is only one step of heat transfer to the cooling air.

IX.2.3.3 Air cleaners

Air cleaners pose a significant challenge for the propulsion unit designers, but as the filtration technology is relatively undeveloped, there are significant potential gains in this area. Incremental improvements have and will continue to be made in the design, medium, and construction of barrier filters. The Donaldson barrier filter in the dAIPS is self-cleaning by means of compressed air supply. This necessitates a high-speed compressor that must be driven through a special gear train. There is an obvious opportunity of space saving and increased reliability by facilitating the self-cleaning for the boost air tapped from downstream of the turbocharger compressor.

Pressure side filtration is the placement of the barrier filter downstream of the turbocharger compressor. This has several significant advantages:

- A given blockage at the pressure side filter causes less engine gas exchange loss (pumping loss) than the same restriction would as a barrier filter upstream of the

compressor. This is because pressure loss over an orifice is inversely proportional to charge density.

- The filter can be made smaller than the conventional barrier filter by a factor of the increase in charge density. The pressure side filter may therefore be less than half the size of the conventional barrier filter.

Pressure side filtration does require efficient prefiltration upstream of the turbocharger compressor. The vortex tubes in the prefilters can remove particles over 5 microns in diameter. These filters are approximately 95 percent effective. It is not clear how fast a compressor wheel will erode when subjected to particles not removed by the prefilter. However, there are various countermeasures to wheel erosion, such as the use of ceramic wheels or ceramic-coated wheels. The life of the combat engine is not as long as that of a heavy-duty commercial engine, so the problem of wheel erosion may be overstated.

Increased efficiency of precleaner vortex will reduce compressor wheel erosion and also permit smaller pressure side barrier filters with longer filter change-out periods. At the forefront of the vortex tube technology is Technology Import Exchange of Redmond, Washington. This company markets vortex tubes that were developed by the South African Atomic Energy Authority. These tubes remove particles over 2 microns in diameter.

It is suggested that pressure side filtration and the necessary associated technologies be investigated and demonstrated by TACOM.

IX.2.3.4 Fans

The fans that are required to cool the propulsion unit consume 10 to 15 percent of the engine power output. In times of burst power demand, the fans are switched off for a period of approximately 1 minute, before the coolant temperature rises to a dangerous level. As far as can be determined, there is no technology that can increase the efficiency of the fans themselves. However, some gains could be made in the efficiency of the fan drive. Direct drives to the fans are the most mechanically efficient but in this case, the fans operate at varying speeds and may supply more cooling air than is actually required. If the fans are electrically or hydraulically driven, the fan speed or the number of fans operating can be controlled to suit the required cooling airflow. If the electrical or hydraulic fan drives can be made reasonably efficient, a comprehensive control system may be the best method of minimizing fan losses.

IX.2.4 Transmissions

IX.2.4.1 Integration

The further integration of the powertrain is the most significant area for further volume reduction in the transmission. Integration in the dAIPS program made great strides in volume reduction of the transmission, but more gains are possible with complete integration of engine and transmission. Currently, there are three walls between the two components plus an air space: the range pack wall in the transmission, the outside wall of the transmission, and the outside wall of the engine. The air space carries away heat. This can be reduced to the range pack wall and

heat carried away by the lubricant. Other components of the power pack can also be better integrated. For instance, a hydraulic cooling pump can be incorporated into unoccupied space in the transmission rather than attached to the outside of the engine or transmission.

IX.2.4.2 Mechanical transmissions

a. Higher speeds

With the possibility of engines going to higher speeds, rather than reduce the speed at the input of the transmission, it may be possible to operate the transmission at higher speeds, thereby reducing the size of the components. Of course, smaller diameter means higher slip speeds, higher stresses, and higher clutch energies. The impact of these changes will be discussed in the sections that follow.

b. Input

Currently, transmissions for diesel engines do not have an input reduction ratio. It would minimize size if the transmission could operate at the speed new reciprocating engines reach. If an input reduction were found to be required, it would not be a problem to establish a proper input speed, since gas turbine transmissions have about eight-to-one reduction ratios already.

c. Torque converter

With an increasing number of transmission ranges, the torque converter has become less important. With the proper electronic controls, it can be eliminated and starting can be accomplished with a controlled clutch engagement. This would be a significant volume reduction for the transmission.

d. Range pack

A reduced size range pack with higher input speeds would have higher clutch slip speeds and higher clutch torques. To do this, a significantly higher energy clutch material must be developed. Some carbon type clutch materials do show promise but to date, no satisfactory bonding material has been found to attach the face material to the core plate.

e. Brakes

Higher speed, smaller diameter brake plates also require a much higher energy material as range plates.

f. Clutch material

The important advance needed for higher speed transmissions is a higher energy clutch plate as described above. This is one of the important research areas for the future smaller size power packs if mechanical transmissions are to be utilized. This research includes new clutch materials and new bonding methods or materials.

g. Lubricant

New high temperature lubricants will be required for smaller power packs. The new lubricant will allow higher operating temperatures, which will reduce the size of the cooling system. The lubricant must also be compatible with both engines and transmissions to allow a common sump. This is also a major research area for smaller power packs.

IX.2.4.3 Electric transmissions

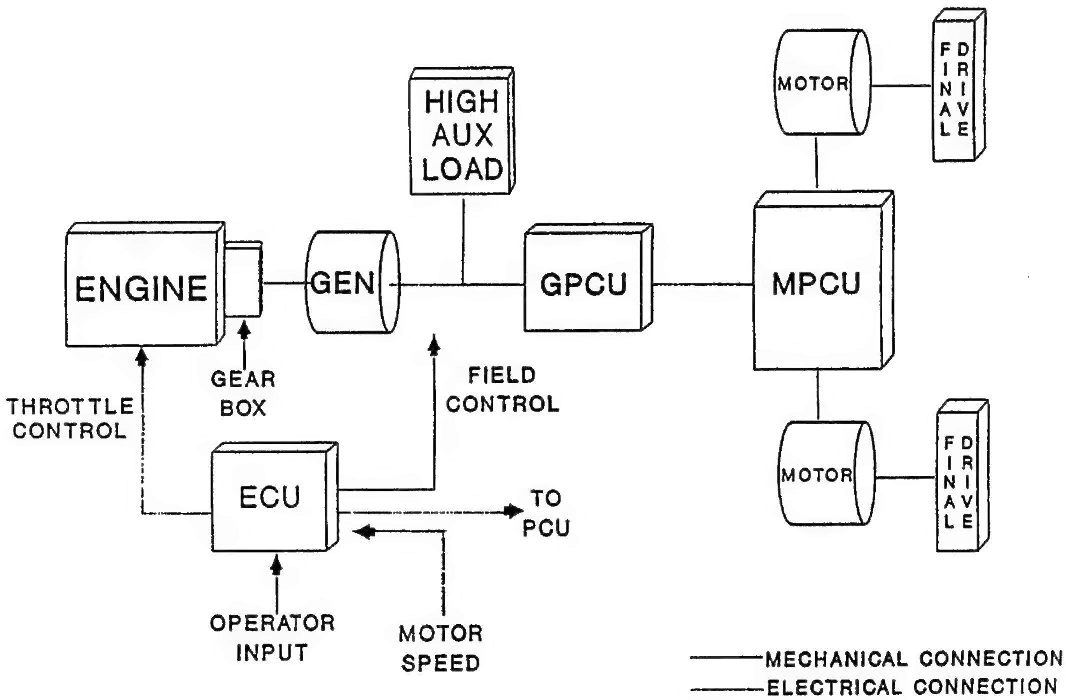
The transmission in a vehicle matches the road load/tractive effort (TE) at a given speed (v) to the prime mover output shaft torque (T) at a given shaft speed (n). If the efficiency of the transmission is denoted by *eff*, then the steady-state power at the prime mover shaft is related to the power expended at the vehicle wheels or tracks by

$$eff \times T \times n = TE \times v. \qquad \text{(Eq. 15)}$$

This power balance equation applies to all types of transmissions since it simply relates the input and output quantities of the particular transmission. The various types of transmissions may or may not impose additional requirements or relationships between these input and output quantities. For example, a pure mechanical or gear transmission forces a linear relationship between the input and output speeds and the near inverse of this linear relationship between the input torque and the output force. This linear coupling between the road speed and the prime mover shaft speed impacts on the design of the prime mover. In the main, due to the requirement of full-power output over a rather large speed range in military vehicles, the engine designer is forced to provide for increasing torque capability as the engine speed decreases in order that near constant shaft power can be generated over a given speed range. This capability is known as the "torque rise" of the engine.

Not all transmissions, however, require torque rise capability in an engine to be able to deliver rated engine power over a range of vehicle speeds. In particular, the electric transmission requires no such capability. In an electric transmission, the engine speed and the vehicle speed are completely decoupled. Thus, the engine designer is freed to optimize engine performance in terms of only power and efficiency, with no restrictions or requirements on low-speed or mid-speed torque capability. An electric transmission is a true, continuously variable transmission.

A generic electric transmission is shown in Fig. IX-20. Engine shaft power is converted to electrical power in an engine-driven generator or alternator. [An alternator is an electrical generator that produces alternating current (AC) output.] This electric power is then conditioned or managed by solid-state switching devices and distributed to traction motors at the vehicle wheels or tracks for reconversion to mechanical power. All modern electric transmissions or drives are AC drives; that is, the currents produced by the engine-driven alternator and the currents driving the traction motors are alternating currents. In general, the frequencies of these currents in the alternator and the motors are different and independent of each other. The power



(Note: Components that have electrical connections can be moved to optimum location.)

Figure IX-20. A generic electric transmission

conversion units, or PCUs, at the alternator output and traction motor inputs adjust and control the level of current required by each machine. This control is coordinated by a master controller denoted as the electronic control unit, or ECU, in Fig. IX-20. Any type of AC electrical machine, such as an induction machine, a wound field synchronous machine, a permanent magnet (PM) field synchronous machine, or a reluctance synchronous machine can be used as the alternator or as the traction motors in an AC drive. A combination used in many recent vintage drives is a synchronous alternator, either wound field or PM field, and squirrel cage induction traction motors. The standard PCU configuration for this particular drive is an alternator-driven rectifier that converts the alternator AC power (an engine speed-determined frequency) to direct current, or DC power, which is then inverted at the traction motor PCUs to AC power at frequencies needed by the motors, which is determined by their output mechanical (vehicle) speed. This type of drive is referred to as a DC link-type drive since there is an intermediate conversion to DC power, which has no frequency, and thus, no shaft speed or vehicle speed associated with it. The alternator frequency, which is engine speed-related, and the traction motor frequency, which is vehicle speed-related, are completely independent. Only the input and output powers are related by the drive efficiency. The speeds of the input and output machines are totally unrelated.

The traction motors in an electric drive are sized (physically sized) by the drive torque requirements at the vehicle wheels or tracks. The engine driven-alternators, however, are sized by the value of the maximum engine output power. Since the magnitude of the voltage produced in an electric generator is proportional to the relative speed at which a magnetic field moves across or through an electric circuit, it is always advantageous, for reducing the size of an

alternator, to operate the machine at high shaft speeds. The higher the speed, the smaller the machine. This is the reason that in many electric drives there is a speed-up gear box between the engine and the alternator. Thus, the need to increase engine shaft speed (or equivalently, engine piston speed) to increase prime mover power density does not present a problem for electric drive systems. In fact, it would be advantageous to do so since it could eliminate the need for the alternator speed-up gear box.

None of the modern (i.e., since the advent of solid-state power electronic devices) electric drive studies or electric drive conversions conducted for the Army have considered the power train as a single system. They were, if hardware, retrofits, or if studies, retrofit studies. They all made use of existing powerplants that were originally designed for mechanical types of transmissions. A "clean sheet of paper" design or study could make use of the continuously variable aspects of the electric drive to free constraints forced on most engine designs due to the speed requirements of a mechanical transmission. In particular, the requirement of a torque rise capability could be totally eliminated. The ramifications of this and other design freedoms should be investigated on a system optimization basis. A study effort considering the impact of the use of electric transmissions on the system optimization of intermittent combustion-powered vehicles is the Committee's primary recommendation for research in the general area of electric transmissions.

For research in a specific area of electric transmissions, work on packaging of power semiconductor devices to enable the use of higher temperature cooling liquids is suggested. At present, power devices such as thyristors (a generic name for semiconductor-controlled rectifiers, or SCRs) and transistors, including the most used and efficient power transistor type, the insulated gate bipolar transistor (IGBT), are packaged in "hockey puck" or module-type units. These individual devices must then be combined and mounted within an electrical connection (mechanical support) cooling assembly, which transfers heat due to device losses to a circulating coolant internal to the mount/assembly, which, in turn, transfers the heat to a heat exchanger or radiator. At present (and at least for the next 10 to 20 years), all power devices are made from crystalline silicon material, which is limited to maximum temperatures of operation, referred to as maximum junction temperatures, of 150°C. Coolant fluid for silicon devices must then be limited to operating temperatures (heat exchanger outlet temperatures) below a safety or derating value of 100 to 125°C, minus the temperature difference required to extract the heat via conduction from the device through the package and mount to the coolant fluid. Currently, there is a research effort to perfect silicon carbide material which if successful, will push maximum junction temperatures to above 300°C. The funding for this work, however, when compared to other expenditures in power device research, is so small that commercial results cannot be expected for at least a decade.

A near-term solution to the excessively low coolant fluid temperatures required for present day packaged power devices and the resultant excessively large radiator surfaces required to reject heat at these temperatures to the military ambient requirement of 120°F is to lower the thermal resistance to conduction heat flow between the silicon device itself and the coolant fluid. A measure of the required performance improvements can be seen from the thermal resistance data shown in Fig. IX-21. It is desirable for each device to be able to continuously operate at heat rejection levels (due to device losses) between 400 to 1,000 watts. State-of-the-art IGBT

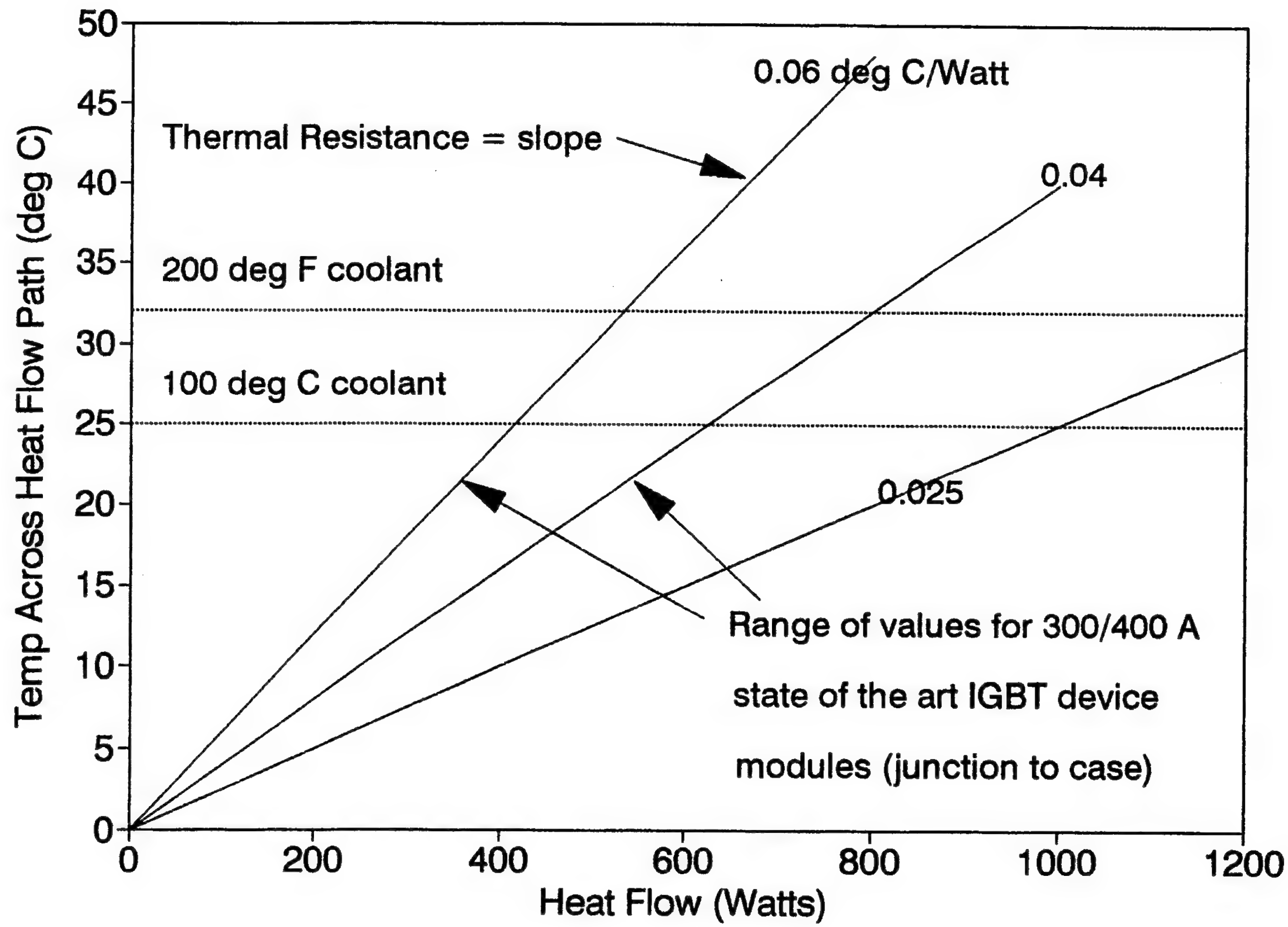


Figure IX-21. Thermal resistance requirements for power device package/mount with 125°C maximum junction temperature

modules have equivalent thermal resistance values between 0.04 to 0.06 C/watt from the device itself to the exterior of the module case at the heat transfer surface. As can be seen from Fig. IX-21, if coolant temperatures near 100°C are desired, values such as these are required for the complete transfer path of device to coolant. Present designs, using state-of-the-art IGBT modules, are forced to limit coolant inlet temperatures to maximum values of approximately 65°C. Raising this temperature to 100°C would reduce the required frontal area of the radiator used to (separately) cool the power electronics by more than a factor of three. Some work in this area is currently underway in the Armored Systems Modernization (ASM) program, but a separate effort to develop a generic packaging/mounting structure for power devices is suggested.

IX.2.6 Propulsion system fluids

IX.2.6.1 Fuels

Several of the special briefing presentations to the BRC mentioned potential fuel concerns for the next generation intermittent combustion engine. The fuel-limiting technology barriers were identified as the following:

1) thermally stable fuel having acceptable lubricity, and
2) compatibility of stable fuel with engine materials.

The Army emphasized that increased heat rejection to the fuel is to be expected. With increased heat soakback at shutdown, the fuel thermal and oxidative stability cannot be taken for granted, especially that of No. 2 diesel fuel. While operating injector tip temperatures are in the neighborhood of 200°C (392°F), surrounding metal temperatures are over 600°C (1,112°F). Aviation turbine fuels currently have a thermal stability requirement at 260°C, which may or may not meet new diesel/rotary engine requirements. Since diesel fuel specification does not have a thermal stability requirement, injector sticking and/or heat exchanger fouling problems are more than likely to prevail in the future engine.

The Army and Marine Corps are faced with unique problems in using fuels of distinctly varying qualities, i.e., fuels that range from low-sulfur, low-aromatic varieties to third-world distillate fuels having high sulfur and high end point. When fielded, the to-be-developed Army high-output compact intermittent combustion engine will be required to operate on and accommodate these different fuels.

Several factors further complicate the fuel issue for the future Army high-output, compact intermittent combustion engines. The first factor is the environmental requirement to reduce fuel sulfur and aromatic content of diesel fuel as a means to control diesel exhaust emissions. Refinery processing to lower the sulfur and aromatics normally removes natural corrosion inhibitors and surface-active compounds in the fuel that provide injection system wear protection. This increased processing is the main reason that Jet A-1/DF-A fuels, which normally have very low sulfur and reduced aromatic content coupled with a viscosity at 40°C one-third to one-half that of No. 2 diesel fuel, have more severe oxidative-corrosive wear tendencies than does diesel fuel. Simply stated, Jet A-1/DF-A fuels have a lower viscosity and are more highly refined and cleaner than diesel fuel. While JP-8 and JP-5 also have lower viscosity and are highly refined

and clean fuels, these fuels require a corrosion inhibitor additive that functions as a lubricity enhancer, which also provides wear protection.

A second factor that will complicate the fuel issue is the expectation that future high-output engines will use higher injection pressures to enable more rapid mixing and improved in-cylinder air utilization. Coupled with the higher heat flows around the combustion chamber, conditions for unit injector distortion, misalignment, and resultant component wear are expected to prevail, especially if lower viscosity, thermally stable clean fuel is used.

A third factor to be considered is the compatibility of the future engine on high-sulfur, high end-point fuels that are widely used in third-world nations. The effects of using these fuels are discussed under the Lubricants Issues.

IX.2.6.2 Lubrication

The consensus among the special briefing presentations was that the future intermittent combustion engine will need to be lubricated with either a new/advanced liquid lubricant in conjunction with durable engine materials (i.e., ceramics, composites; monolithic and coated), or with a novel lubricant/lubrication system. There will be a need for challenging research in the engine lubrication area, further complicated by unique military and commercial lubricant requirements. For example, the Army would like fewer lubricants on the battlefield instead of adding a new, special, high-temperature lubricant. Further, very stringent diesel engine exhaust emission reductions underway in the 1990s will focus on lubricant development that is driven by low-sulfur, low-aromatic fuels. Along with the to-be-developed Army high-temperature lubricant, these low-emission engine lubricants will need to be compatible with already fielded Army engines and power transmissions of the 1980s and 1990s, using the broad range of fuels expected to be available (i.e., from low-sulfur, low-aromatic fuels in the U.S. to the high-sulfur, high end-point fuels in third-world countries).

dAIPS Conditions-Current Baseline – It was generally agreed that the lubricants baseline to improve upon is the temperature regime in the current TACOM Advanced Integrated Propulsion System (dAIPS) engine. These temperatures and operating goals are as follows:

Top Ring Reversal	300°C (572°F)
Bulk Oil	171°C (340°F)
Oil Drain Interval	300 hours; 3,000 miles.

For the next generation compact/high-output intermittent combustion engine using an advanced synthetic base and additive system, the oil temperatures and operating goals may be estimated as follows:

Top Ring Reversal	345°C (650°F)
Bulk Oil	190°C (375°F)
Oil Drain Interval	400 hours; 4,000 miles.

In the initial stages of dAIPS engine development, 595°C (1,100°F) top ring reversal temperatures were targeted.[1] Due to a lack of suitable liquid lubricants being available, this temperature level has not been achieved with any reasonable reliability in screener or full-scale demonstrator engines. Accordingly, the current generation dAIPS lubricant temperature goals shown above as 300°C (572°F) top ring reversal and 171°C (340°F) bulk oil temperature have been achieved using synthetic polyolester-based lubricants. The areas of dAIPS oil performance that need improvement include increased piston cleanliness; reduced wear of piston pin bushing, main and rod bearings, and overhead components; and reduced oil consumption.

It is judged that synthetic liquid lubricants capable of operation to 345°C (653°F) top ring reversal and 190°C (374°F) bulk oil temperatures will likely be available for the next generation military compact/high-output intermittent combustion engine. A breakthrough in synthetic lubricant technology might allow the following temperature goals as claimed in the recent report by Sutor, 1993:

Top Ring Reversal	538°C (1,000°F)
Bulk Oil	260°C (500°F)
Oil Drain Interval	500 hours; 5,000 miles.

As shown in Fig. IX-22, the engine cooling system volume reduction is estimated to be up to 8 percent if the bulk oil (sump) temperature can be elevated from 171°C (340°F) for the current

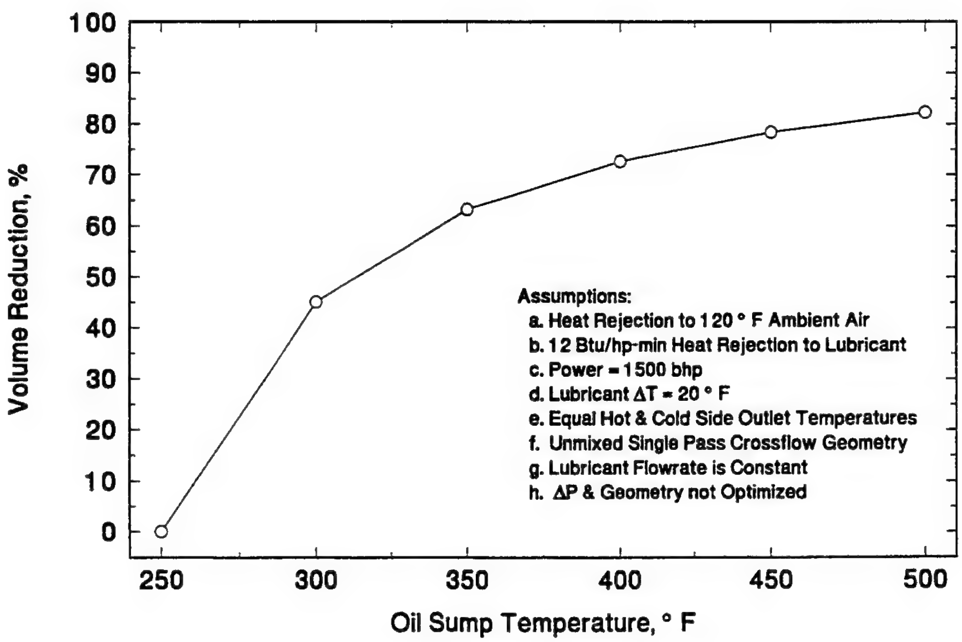


Figure IX-22. <u>Heat exchanger volume reduction (heat transfer to 120°F ambient day)</u>

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[1] Bryzik, 1978; Bryzik and Kamo, 1983; Frame, 1983, 1992; Kamo and Bryzik, 1978, 1981; Kanakia and Peterson, 1986; Kanakia, Owens, and Peterson, 1985; Radovanovic, Kamo, and Dufrane, 1983; SAE Progress in Technology, 1984; SAE SP-543, 1983; SAE SP-571, 1984; SAE SP-610, 1985; Stang and Johnson, 1977.

AIPS to 190°C (374°F) for the future compact high-output intermittent combustion engine, or up to 22 percent if 260°C (500°F) bulk oil temperature is achieved.

A summary of approximate temperature capability for diesel and turbine engine lubricants is provided in TABLE IX-10. The upper temperature use limits have been established from either military practice, estimates from laboratory research, or from thermal decomposition measurements. In diesel engines in which plenty of oxygen is available from combustion products and blowby gases, the lubricant will degrade at temperatures well below the thermal decomposition temperatures shown. However, if the same piston temperatures were possible, stoichiometric diesel combustion should help oxidative stability due to less oxygen being available to decompose the lubricant.

TABLE IX-10. Approximate Temperature Range Capability for Turbine and Diesel Engine Lubricants

Lubricant Class (MIL-SPEC)	Approximate Temperature Range* °C	Approximate Temperature Range* °F
Mineral Oils (MIL-L-2104)	–25 to 122	–15 to 250
Polyalphaolefins, PAO-Diester Blends (MIL-L-46167)	–55 to 122	–60 to 250
Polyolester (MIL-L-7808; MIL-L-23699)	–55; –40 to 200	–60; –40 to 400
Improved Polyolester (MIL-L-27502)	–40 to 220; 340†	–40 to 428; 645†
Aromatic Ester	340†	645†
Perfluoralkylether	390†	735†
Polyphenylether (MIL-L-87100)	–60 to 300; 443†	–60 to 572; 830†

* Upper limit is maximum continuous bulk oil temperature.
† Thermal decomposition temperature (Jones, W.R., "Tribology in the 80's," NASA Conference Publication 2300, Section 4, p. 419, April 1983).

As the search continues for improved synthetic liquid lubricants capable of operation up to 700°F (or above), serious consideration should be given to the concurrent use of decomposition products and/or solid films that may work in critical areas. The major concerns in the use of these solid

films/decomposition products will be their frictional, wear, and stability characteristics at temperatures above 371°C (700°F), while still being effective at lower temperatures. Efforts should focus on finding effective "residual" lubricant films deposited from synthetic fluids, which consist of decomposition films, surface active additives, and solid lubricants. The goal would be to extend the operating temperature as high as possible without jeopardizing fluid film or boundary lubrication at lower temperatures. The residual film should lubricate both as a decomposition film and when that is no longer effective, leave a solid lubricant film. Mixtures of graphite and soft metal oxides might be considered for solid film lubricant (Kanakia, Owens, and Peterson, 1985).

<u>Materials</u> – As the lubricants have been improving over the years, so have the engine materials. From all indications, the ring and liner area still needs some form of lubrication that will allow full-engine operational performance and reasonable durability. Ceramics/composite materials have the advantage of insulation, high hardness, maximum utility during marginal lubrication, and even the ability to operate unlubricated for short periods of time. The future Army compact high-output intermittent combustion engine will require both improved materials and advanced liquid lubricants (synthetics).

<u>Two-Stroke Versus Four-Stroke Cycle Diesel</u> – In the course of developing lubrication requirements for the future compact high-output diesel, it was recommended that the BRC revisit the compound cycle engine work conducted at Garrett Engine Division for the Army Aviation Systems Command (AVSCOM—NASA) in the 1980s. In that work (Castor, 1981; Castor, Cruse, Eckelberg, and Martin, 1988, 1989), a specially modified Detroit Diesel, Series 53, Uniflow two-stroke cycle diesel engine and an AVL loop scavenged single-cylinder engine were used to investigate basic engine power, performance, heat flows, lubrication friction, and wear rates. Technology was conducted at power densities up to 317 IMEP at 6,000 rpm in the loop engine and 300 IMEP at 4,000 rpm in the uniflow engine, all with normal water jacket cooling and a special synthetic liquid lubricant.

The Garrett research demonstrated high-speed, high-BMEP capabilities and on-line generation of piston ring wear data. However, the top ring reversal temperatures in the range of 281 to 295°C (538 to 563°F) and oil sump temperatures only in the range of 102 to 109°C (216 to 228°F) did not exceed dAIPS lubricant temperatures, since the engines were run without minimal cooling.

The future high-output diesel employing the traditional two-stroke cycle concept would present a major lubricant challenge from the standpoint of piston/cylinder cooling and heat rejection. The characteristic shorter cooling period compared to that of the four-stroke cycle diesel would make it extremely difficult to cool the two-stroke piston/cylinder liner area without cooling jackets containing liquid coolant at relatively high flow rates. Extensive research would be required to determine if lower heat rejection benefits could be achieved through dual use of engine lubricant as the jacket coolant and the piston coolant. However, a lubrication benefit in the two-stroke process is that while there is not much time to cool the piston/liner area, there also is not much time to degrade the bulk lubricant, which is more likely to occur in the four-stroke diesel. The two-stroke cycle diesels have higher brake specific oil consumptions than their four-stroke counterparts; hence, less degraded lubricant from the ring zone area finds its way into the bulk

oil sump. Hot ring sticking or ring breakage accelerates two- or four-stroke engine lubricant degradation in the form of viscosity and acid number increases.

Critical wear process in the ring/liner area of both two- and four-stroke engines depends on deposit control and maintaining piston-to-liner clearances. Deposit buildup in ring grooves and on piston lands contributes to ring sticking, lubricant transport to the upper ring zone area, reduced cooling, ring and liner scuffing/wear, and ultimate failure. Crown land and ring land deposits, along with poor piston cooling, lead to reduced clearances that cannot be tolerated in a very high-output diesel without rapid failure. In a nondeposit-related process such as found in a relatively new engine, it is obvious that either low oil or coolant levels cause a loss in piston-to-liner clearances, increased ring wear, loss of oil control, and ultimate seizure. A real-world example of this is the fact that the Army fleet of turbocharged two-stroke, 2,800-rpm diesels fitted with trunk pistons will not cool the piston/liner area if operated with conventional SAE 10-grade mineral oils. This lack of cooling results in reduced life and potential rapid piston-to-liner seizure (Lestz, 1973; Lestz and Bowen, 1975; Stavinoha, Eichelberger, Lestz, and Tyler, 1980). In cold climates, special arctic engine oil is specified for which qualification performance testing requires use of a much less volatile/synthetic base product (diester, polyalphaolefin, etc.). Four-stroke diesels in the Army inventory operate acceptably with either the mineral or synthetic winter grade oils, demonstrating far less sensitivity to lubricant volatility and viscosity. A two-stroke diesel is viewed as having less tolerance for marginal cooling conditions mainly due to the characteristic shorter time in the cycle for cooling/heat rejection; hence, this reduced time is the apparent cause for viscosity and volatility sensitivity even with full cooling in the water jackets. However, significant improvements have been made in military two-stroke diesel heat rejection and cooling. Recently, the Army introduced an improved, low-heat rejection 8V-71T diesel engine for use in the M109 and M992 self-propelled artillery and ammunition support vehicles. This engine makes use of ceramic coatings in key cylinder areas and is fitted with cross-head pistons and keystone-grooved, barrel-faced rings (replacing the trunk pistons and rectangular-grooved, rectangular-faced rings). These improvements result in reduced cooling requirements, increased power output, and improved lubrication/reduced ring and liner wear (Freese, Hinkle, and Avery, 1993).

The difference in two-stroke versus four-stroke cooling time is also important in the engines' fuel sulfur and lubricant ash level tolerances. The four-stroke diesel is more tolerant of fuel-sulfur levels above 0.50 wt% and lubricant sulfated ash levels above 1.0 wt% than is the two-stroke diesel. The two-stroke fuel-sulfur and lubricant-ash intolerance often causes early ring sticking, ring and liner wear, and exhaust valve burning. The typical four-stroke diesel is not nearly as sensitive and can run continuous hours on high-sulfur fuel, employing more frequent oil drains that are dependent on the new and used oil chemistry (alkalinity content). Overall, engine life is reduced in either engine class even with more frequent oil drains that are directly related to the fuel-sulfur and lubricant reserve alkalinity levels (Lestz, LePera, and Bowen, 1976). Also, higher sulfur fuels often have higher end-point distillation temperature that leads to rapid piston/ring zone and injector tip deposits in high-speed diesel engines. However, the two-stroke, high-speed diesel is less tolerant of the higher end-point fuels than is its four-stroke counterpart.

<u>Novel Lubricant Delivery Systems</u> – An earlier study (Kanakia, Owens, and Peterson, 1985) presented several potential lubricant delivery systems for high-temperature use. These potential systems are shown in TABLE IX-11. The study concluded that some of these delivery systems

were not viable alternatives in long-term applications. In particular, regardless of the materials and lubricant selected, a continuous liquid lubricant delivery system is necessary, i.e., exclusive use of solid lubricant coatings and composite materials is not currently possible. Since changing the lubricant delivery system in the cylinder would require major redesign, every effort should be made to keep the current crankcase lubricant delivery system. This design might be retained by exploiting the "residual lubricant" concept mentioned above. Active lubricant suspended in a carrier liquid or gas shown as Items 3 and 4 in TABLE IX-11 should be considered. As noted, in this system an active lubricating compound is suspended in a once-through or recirculating carrier fluid, which evaporates from the hot surface. The "carried lubricant" remains on the hot surface as a solid film to reduce friction and wear. Since benefits of using solid film lubricants accrue at elevated temperatures, it may be best to isolate their use to the hottest section of the engine as in the upper ring zone. This system would result in use of the separated system (Item 11 in TABLE IX-11) in much the same manner as is done in rotary combustion engines and in large bore, slow-speed cross-head marine diesel engines. Other lubricant delivery systems or combinations should also be considered, including use of high-temperature lubricants, compatible in both the engine and transmission, utilizing a <u>common</u> sump.

IX.2.6.3 <u>Fuels and lubricants summary requirements</u>

Future military high-output compact intermittent combustion engine development will need to address the integration of the following:

- fuels stability and compatibility;
- advanced synthetic lubricants–base stocks and additive systems;
- advanced engine materials (monolithics and coatings); and
- novel lubricant delivery systems.

IX.2.6.4 <u>Suggested research in the fuels area</u>

Specific research needed to address the <u>fuel issue</u> includes the following:

1) thermal flux measurements and modeling of temperature environment in and around the fuel injection system;

2) investigation of the effect of fuel physical-chemical property variables on high-temperature lubricity and deposit formation in the fuel injection system, to include viscosity, volatility, density, and chemical composition;

3) methods of on-board fuel treatment/pretreatment to reduce fuel system deposit formation; and

4) investigation of candidate fuel-engine-materials compatibility at high temperatures, to include injection system wear, corrosion, seals, and filters.

TABLE IX-11. Potential Lubricant Delivery Systems for High-Temperature Use

System	Description
1. Synthetic Lubricants	Conventional hydrocarbons are replaced by more stable high-temperature synthetic fluids along with high-temperature materials. Current lubricant delivery techniques are used.
2. Decomposed Fluid	A once-through lubrication system provides a fluid lubricant to a hot surface. The decomposing fluid, both liquid and solid, is an adequate lubricant.
3. Liquid-Carried Suspension	A liquid is used as either a "once-through" or recirculating lubricant that contains the solid lubricant or the ingredients of the solid lubricant in suspension. Evaporation or decomposition of the liquid at operating temperatures deposits the solid lubricant.
4. Gas-Carried Suspension	Similar to Item 3 except that a gas is used as a carrier.
5. Unlubricated	Piston ring and liner materials that will meet friction and life requirements without being lubricated are chosen.
6. Solid-Film	Lubricating films are originally applied to ring and liner materials but do not require resupply.
7. Composite Materials	Materials are used that contain lubricant to reduce friction, wear, and surface damage.
8. Reactive Gases	A gas is supplied to the component that reacts with the surface to form a lubricating film.
9. Stick Transfer Films	A solid stick of lubricant is rubbed against a surface to supply a lubricant film.
10. Sacrificial Stick System	A stick of solid lubricant is inserted into or placed adjacent to a ring or bearing surface. As the bearing surface wears, lubricant is applied.
11. Separated System	Consumable lubricant used in isolated ring zone area; recirculating liquid lubricant used in lower end and elsewhere.
12. Segregated System	Lower end and cylinder lubrication with highly stable base stock and additives; segregated overhead (camshaft, gearing, etc.) with "conventional lubricant" and additives.

IX.2.6.5 Suggested research in the lubrication area

Specific research needed to address the future powerplant lubrication issue includes the following:

1) develop optimized synthetic base stock chemistry that also has good high- and low-temperature viscosity characteristics;

2) define base stock-additive compatibility-solubility;

3) develop improved high-temperature additive components, such as antioxidant, antiwear, and detergent-dispersancy;

4) investigate latest advanced materials and surface-treating processes (i.e., ion implantation, etc.);

5) investigate synthetic lubricant versus engine materials compatibility (i.e., wear and corrosion in lower-end bearings, cam/valve actuation wear, and seals); and

6) define trade-offs of lube requirements versus engine design using a novel, separated, or segregated lubricant system such as recirculating lubricant in the lower-end bearings, gears, etc.; a once-through lubricant in the ring zone/combustion area; and/or common sump for engine and transmission oil.

IX.3.9 Volume and weight estimation methodology

The BRC had requested that TACOM conduct propulsion system volume and weight analyses for the advanced intermittent combustion engine compact systems addressed within this report and their overall impact on vehicle system weight. Results and methodology behind the analyses are presented within this section. It should be noted that the volume numbers for the 60-ton, 1,500-hp case presented here are TACOM estimates and do not directly correspond to the BRC numbers in Volume I, Appendix 9.3.9. The TACOM number for the entire propulsion system volume for this case is, however, within 3 percent of the BRC estimate. This 3-percent difference between the TACOM vs. BRC estimates is not considered significant. Weight estimates were correspondingly made using the TACOM volume estimates. Volume and weight estimates are provided for a 30-ton vehicle, 750 hp and 1,000 hp (750 hp mobility plus 250 hp auxiliary) cases and for a 60-ton vehicle, 1,500-hp case, in TABLES IX-12 through IX-19.

IX.3.9.1 30-ton vehicle propulsion volume estimates

The following criteria were used for component volume estimates for both the scaled dAIPS and the Advanced dAIPS cases for the 30-ton vehicle:

A. Engine – Directly proportional to engine displacement.

B. Transmission – Three-quarters directly proportional to gross vehicle weight; one-quarter directly proportional to engine gross horsepower.

C. Cooling – Directly proportional to heat rejection.

D. Air Filtration – Directly proportional to engine airflow.

E. Exhaust System – Directly proportional to engine airflow.

F. Fuel System – Directly proportional to vehicle ton-miles per gallon.

G. Batteries – Directly proportional to engine displacement.

H. Unused – 10 percent of total used volume.

The baseline from which the 750-hp and 1,000-hp scaled dAIPS volumes were estimated was the current dAIPS 1,500-hp engine. The 1,500-hp engine has a space claim of 34 cu. ft, and the total propulsion system space claim associated with that engine is 170 cu. ft. The advanced dAIPS volume numbers are projections based on engine technology improvements over the scaled dAIPS versions which are expected to occur by the year 2005. Improvements in engine specific air consumption, specific heat rejection, and specific fuel consumption positively impacted space claim projections for the advanced improved dAIPS engine volume data presented in Volume I, Appendix 9.3.9. Transmission volume space remained constant in a given bhp class in this study. Nevertheless, advancements in transmission technology resulting in reduced weight and volume can be expected by the 2005 timeframe.

IX.3.9.2 Propulsion systems weight estimates

The baseline for Volume I, Appendix 9.3.9 weight estimates is the dAIPS 1,500-hp engine. Engine weight is 4,170 lb, and the total system weight associated with that engine is 14,565 lb. Components and criteria for estimates included in the weight study include those items in categories A through G given in Section IX.3.9.1. Those criteria as used in the volume analysis were used to determine weight (volume considered directly proportional to weight) of the 750 hp and 1,000 hp scaled dAIPS engines. Final drives were not included in the weight analysis. The weight estimate for the advanced diesel engine for the 60-ton vehicle was based on previous estimates where a 25 percent reduction in engine weight was deemed achievable. Weight reductions were attributable to technology combinations of increased engine speed, operation at lower air-to-fuel ratios, and use of lightweight, high strength ceramic, composite and metal alloys throughout the engine design where possible. Transmission weight reductions were obtained via vehicle weight reductions due to a decrease in overall propulsion system weight. An iterative step was used to determine this transmission weight reduction. The same type of procedure was followed in fuel system estimates whereby fuel system weight savings were generated by an improvement in ton-miles per gallon via lower engine specific fuel consumption and by the propulsion related vehicle weight savings. Cooling system weight reductions were based on projected lower engine heat rejection levels and air filtration and exhaust system reductions were based on better engine specific air consumption. Battery weights were held constant within the same given power class, but varied as a function of engine displacement between the different power levels. The two reduced weight 750 hp and 1,000 hp cases for the 30-ton vehicle were then scaled from the 60-ton case using those criteria A through G in Section IX.3.9.1.

IX.3.9.3 Vehicle system weight estimates – variable vehicle weight and hp/ton

Six weight comparisons were made to assess propulsion system volume reduction impact on total vehicle system weight. Propulsion system volume comparisons made for vehicle configurations of 60-ton, 1,500 hp; 30-ton, 750 hp; and 30-ton, 1,000 hp can be found in TABLES IX-12, IX-13, and IX-14, respectively. The dAIPS derivatives and "advanced diesel" propulsion system volume reductions vs. the dAIPS baseline form the basis for determining the vehicle system weight reductions. In each case, the level of protection, the hull interior cross-sectional area where the propulsion system is installed, and the gross horsepower were held constant. The hull weight savings for the 60-ton case and the 30-ton case are presented in TABLES IX-15 and IX-16, respectively. The methodology and sample calculations for estimating the hull weight savings are also presented. The savings numbers of 172.2 lb/in. for the 60-ton vehicle and 100 lb/in. for the 30-ton vehicle were obtained from the TARDEC, Advanced Concepts Division, AMSTA-Z. Propulsion weight savings were then combined with the hull savings to show the total vehicle weight savings in TABLES IX-17 through IX-19 for the given vehicle configurations.

IX.3.9.4 Vehicle system weight estimates – constant hp/ton

Section 3.5.3, Volume I contains a discussion (and summary TABLE 3-3) of vehicle system weight savings at constant hp/ton. TABLE IX-20 presents details and methodology for determining the vehicle system weight savings.

TABLE IX-12. Propulsion System Volume – Vehicle Study Assumptions (60-ton, 1,500 hp)

1. Keep level of protection constant.
2. Keep interior cross section area constant.
3. Keep gross power requirement constant.

Case 1 AIPS Diesel Baseline – 1,500 hp 60-Ton Vehicle

Engine	34.0
Trans	34.9
Cooling	16.4
Air Filter	8.4
Exhaust	2.0
Fuel Tanks	39.3
Batteries/Misc.	18.0
Unused	17.0

TOTAL	170.0 cu. ft

Case 2 Advanced Diesel Propulsion – 1,500 hp 60-Ton Vehicle

Engine	18.0
Trans	34.9
Cooling	10.0
Air Filter	5.1
Exhaust	1.5
Fuel Tanks	36.0
Batteries/Misc.	18.0
Unused	13.7

TOTAL	137.2 cu. ft

TABLE IX-13. Propulsion System Volume – Vehicle Study Assumptions (30-ton, 750 hp)

1. Keep level of protection constant.
2. Keep interior cross section area constant.
3. Keep gross power requirement constant.

Case 3 AIPS Diesel Baseline – 750 hp
30-Ton Vehicle

Engine	17.0
Trans	17.5
Cooling	8.2
Air Filter	4.2
Exhaust	1.0
Fuel Tanks	19.7
Batteries/Misc.	9.0
Unused	8.5

TOTAL	85.1 cu. ft

Case 4 Advanced Diesel Propulsion – 750 hp
30-Ton Vehicle

Engine	9.0
Trans	17.5
Cooling	5.0
Air Filter	2.5
Exhaust	0.8
Fuel Tanks	15.8
Batteries/Misc.	9.0
Unused	6.6

TOTAL	66.2 cu. ft

TABLE IX-14. Propulsion System Volume – Vehicle Study Assumptions (30-ton, 1,000 hp)

1. Keep level of protection constant.
2. Keep interior cross section area constant.
3. Keep gross power requirement constant.

Case 5 AIPS Diesel Scaled – 1,000 hp
750 hp + 250 hp Auxiliary Loads
30-Ton Vehicle

Engine	22.7
Trans	18.9
Cooling	10.9
Air Filter	5.6
Exhaust	1.3
Fuel Tanks	26.2
Batteries/Misc.	12.1
Unused	10.5

TOTAL	108.2 cu. ft

Case 6 Advanced Diesel Propulsion – 1,000 hp
750 hp + 250 hp Auxiliary Loads
30-Ton Vehicle

Engine	12.0
Trans	18.9
Cooling	6.7
Air Filter	3.3
Fuel Tanks	21.1
Exhaust	1.1
Batteries/Misc.	12.1
Unused	8.3

TOTAL	83.5 cu. ft

TABLE IX-15. 60-ton Hull Weight Savings Study

(Based upon propulsion system volume reduction)

Assumptions: M1 Configuration Incorporating Future Tank Threat
Armor
2-in. clearance on top
1-in. clearance on sides/bottom

Propulsion Sectional Area
Height: 45 in.
Width: 78.5 in.
Area = 24.5 ft^2

1. AIPS Diesel – 1,500 hp
   Volume = 170.0 cu. ft
   Propulsion Related Hull Length = 83.3 in.

2. Advanced Diesel – 1,500 hp
   Volume = 137.2 cu. ft
   Propulsion Related Hull Length = 67.2 in.
   Length = 83.3 - 67.2 = 16.1 in.

Hull Weight Savings:

Top	56.6 lb/in.
Sides/Skirt	75.9
Floor	9.7
Track	30.0

TOTAL	172.2 lb/in.

Total Hull Weight Savings = 172.2 × 16.1 = 2,772 lb

TABLE IX-16. 30-ton Hull Weight Savings Study

(Based upon propulsion system volume reduction)

Assumptions: 2-in. clearance on top
2-in. clearance on sides/bottom

Propulsion Sectional Area
Height: 45 in.
Width: 63 in.
Area = 19.7 ft^2

1. AIPS Diesel Derivative – 750 hp
   Volume = 85.1 cu. ft
   Propulsion Related Hull Length = 51.8 in.

2. Advanced Diesel Propulsion – 750 hp
   Volume = 66.2 cu. ft
   Propulsion Related Hull Length = 40.3 in.
   Length = 51.8 - 40.3 = 11.5 in.

Hull Weight Savings:

Top	20.0 lb/in.
Sides	32.4
Skirts	29.6
Floor	4.0
Track	14.0
TOTAL	100.0 lb/in.

Total Hull Weight Savings = 100.0 × 11.5 = 1,150 lb

TABLE IX-17. Weight Study for 60-ton, 1,500-hp Vehicle

	AIPS	Advanced
Engine	4,170	3,130
Trans	3,900	3,705
Cooling	580	300
Air Filter	180	110
Exhaust System	50	40
Fuel	2,113	1,776
Batteries	480	480
TOTAL	11,473	9,541

Weight Savings	
Adv Propulsion	1,932
Hull, Track	2,772
TOTAL	4,704 lb

TABLE IX-18. Weight Study for 30-ton, 750-hp Vehicle

	Scaled AIPS	Advanced
Engine	2,085	1,560
Trans	1,950	1,857
Cooling	290	177
Air Filter	90	54
Exhaust System	25	20
Fuel	1,057	808
Batteries	230	230
TOTAL	5,727	4,706

Weight Savings	
Adv Propulsion	1,021
Hull, Track	1,150
TOTAL	2,171 lb

TABLE IX-19. Weight Study for 30-ton, 1,000-hp Vehicle

	Scaled AIPS	Advanced
Engine	2,780	2,085
Trans	1,950	1,833
Cooling	387	236
Air Filter	120	72
Exhaust System	33	27
Fuel	1,409	1,063
Batteries	230	230
	---------	---------
TOTAL	6,909	5,546

Weight Savings	
Adv Propulsion	1,363
Hull, Track	1,320

TOTAL	2,683 lb

TABLE IX-20. Vehicle System Weight Comparison at Constant hp/ton

	AGT 1500	dAIPS		BRC	LCTE*
Horsepower	1,500	1,500	1,300	1,500	1,250
	Weight, lb				
Engine	2,500	4,170	3,614	3,130	2,608
Trans	4,350	4,000	3,467	3,705	3,087
Cooling	596	580	460	300	250
Air Filter	350	165	156	110	91.7
Exhaust	170	46	39.9	40	33.7
Fuel	3,900	1,912	1,701	1,776	1,480
Batteries	480	480	416	480	400
TOTAL, lb	12,346	11,353	9,854	9,541	7,950
Prop sys volume, cu. ft	291	170	147.3	133	110.8
Prop sys wt decrease, lb			2,492		4,396
Veh length reduction, ft†			5.87		7.36
Veh wt decrease, lb‡			12,120		15,199
Total wt decrease, lb§			14,612		19,595
New veh wt, tons♦	60 (Ref.)		52.7		50.2
hp/ton	25.0		24.67		24.9
% wt decrease★			12.2		16.3

* Lowest Comparable Technology Estimate
† (Prop sys vol decrease)/(Prop sys cross section area, 24.5 ft^2)
‡ [(Prop sys vol decrease)/24.5] × 12 × 172.2
§ Prop sys wt decrease + Veh wt decrease
♦ [120,000 - (Total wt decrease)]/2,000
★ [60 - (New veh wt)]/60

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